MACHINERY.

November, 1905.

FANS.—2.

CHARLES L. HUBBARD.

Having taken up the centrifugal fan from a theoretical standpoint and noted its action under ideal conditions, we will now consider it when working under the requirements of actual practice, and show what corrections must be made to the various rules and formulas previously given.

General Proportions.

The general form of a fan wheel is shown in Fig. 11, which represents a double spider wheel with straight blades. Those over 4 feet in diameter usually have two spiders, while fans of

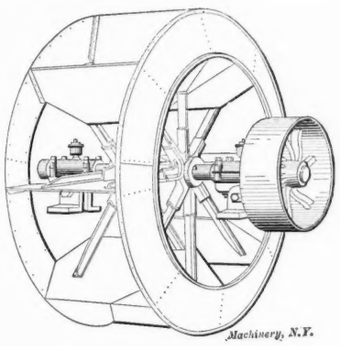


Fig. 11. Construction of Fan Wheel.

large size are often provided with three or more. The number of blades or floats commonly varies from six to twelve, depending upon the size of fan. They are made both curved and straight; the former, it is claimed, run more quietly, but if curved too much will not work so well against a high pressure as the latter form. Fig. 12 represents a section through a fan wheel and shows the principal dimensions to be considered.

The following proportions are averages taken from fans of different sizes as made by several manufacturers for general ventilating and similar work and will be found to vary slightly from the proportions given by any one maker.

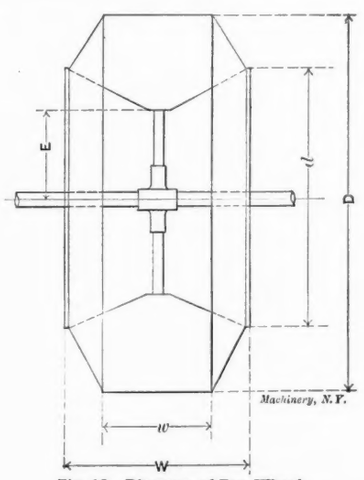


Fig. 12. Diagram of Fan Wheel.

The diameter of the inlet (d) usually varies from .66 to .7 of the diameter of the wheel (D), .68 has been used in the following tables as a fair average. The distance from the center to the heel of the blades (E) is generally made .25 of the diameter of the wheel.

The width W varies somewhat in fans designed for different purposes. In the makes examined it averaged from .50 to .54 of the diameter, and .52 has been used in the tables following.

The width at the periphery should be, theoretically, such that the area of outlet around the entire wheel will be the same as the sum of the openings between the blades at the inlet, but in actual practice it is made somewhat greater, averaging from .7 to .8 of the width of the wheel. For convenience the relations between the different parts of the wheel may be expressed by the following equations:

$d = .68\,D$
$E = .25\,D$
$W = .52\,D$
$w = .8\,W$

in which

D = diameter of wheel,
d = diameter of inlet to wheel,
E = distance from center of fan to heel of blades,
W = width of fan at inlet,
w = width of fan at periphery.

These proportions, as already stated, do not represent those of any particular make, nor follow any fixed rule, but are general averages as found from the catalogue dimensions of several well-known manufacturers.

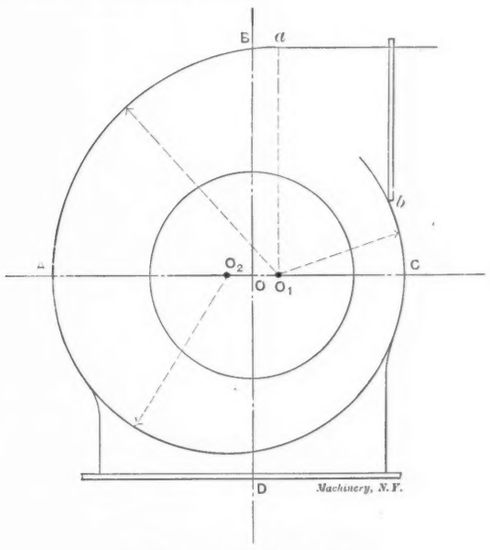


Fig. 13. Layout of Scroll Casing.

Fans are made both with double and single inlets; the former being called "blowers" and the latter "exhausters."

The dimensions of the casing or housing vary somewhat with different makers and can be obtained from their catalogues. The dimensions given in the following tables are taken from the catalogue of the B. F. Sturtevant Co. and will be found useful in the design of ventilating systems in approximating the space required. In case any particular make of fan is to be used, the exact dimensions should be obtained from the manufacturers, especially if the available space is limited.

The scroll of the casing is usually made up of the arcs of three circles of different radii, and the following method will be found useful in laying out a fan casing to scale. (See Fig. 13.)

First draw the center lines and lay off the distances OA, OB, OC and OD from the catalogue dimensions of the fan to be used. Then with a radius R, equal to OB, and a center at O, draw the arc Aa; with the same center and a radius R_1 equal to O_1C draw the arc bC; lay off $O\,O_2 = O\,O_1$ and with a radius R_2 equal to O_2A and a center at O_2 draw the arc AC. The other bounding lines can then be drawn in from dimensions taken

from the catalogue table. The width of the casing is practically the width of the wheel with a small allowance for clearance.

Form of Orifice.

The form of the opening through which air passes when under pressure has a certain effect upon the quantity discharged, and makes it less than the theoretical amount which the size of opening and difference in pressures would indicate.

This reduction is due to two causes. First, to a contraction of the stream within, or just beyond the opening, depending upon its form, and second, to a certain amount of friction which tends to reduce the velocity somewhat.

The ratio between the actual quantity of air discharged and the theoretical quantity is called the "coefficient of discharge" and may be taken as about .8 for the short outlet from a fan casing, and as .56 for the inlet. Example: The pressure within a fan casing is ½ ounce per square inch, and the area of the outlet is 4 square feet. What will be the actual discharge in cubic feet per minute?

From Table I. (October MACHINERY) the velocity corresponding to a pressure of ½ of an ounce is found to be 3,653.8 feet per minute, which gives a theoretical discharge of 3,653.8 × 4 = 14,615.2 cubic feet per minute. To find the actual quantity we must multiply this result by the coefficient of discharge, which gives us 14,615.2 × .8 = 11,692.16 cubic feet per minute.

The quantity of air discharged through any given opening, divided by the velocity of flow, is called the *effective area*. In the preceding example 11,692.16 ÷ 3,653.8 = 3.2 square feet, which is the *effective area* of the outlet, while 4 square feet is th actual area.

Sometimes the conditions of the problem are such that it is stated as follows:

The pressure within a fan casing is ½ ounce, and the outlet is 4 square feet, how much must the pressure be increased to make the actual discharge equivalent to the theoretical at the original pressure? It can be shown mathematically that the ratio of the theoretical pressure to the actual pressure required is $\frac{1}{K^2}$ in which K is the coefficient of discharge. Taking this as .8 for a fan outlet, the ratio becomes $\frac{1}{.64} = 1.56$.

In the above example the theoretical discharge is 3,653.8 × 4 = 14,615.2 cubic feet per minute, in which it is assumed that 4 square feet is the effective area of the outlet. In order to make the actual discharge equal the theoretical, using the same sized outlet, it will be necessary to increase the pressure by 1.56, which gives us ½ × 1.56 = .78 of an ounce.

Blast Area.

While the blast areas of fans of different diameters are slightly different for varying proportions, the formula $\frac{D\,w}{3}$ applied to standard fans will be found sufficiently accurate for ordinary use, in view of the approximations which must be made later in the assumption of pressures to be operated against, due to the friction of the air in ducts and flues.

Asuming $w = .8\,W$, and substituting for W its equivalent, $.52\,D$, we have for the blast area $A = D \times .8 \times .52\,D \times .33 = .14\,D^2$.

TABLE V. DIMENSIONS OF FULL HOUSING FANS.

Size of Wheel.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P
2½	18¾	20½	23½	21	25	24	28⅛	27	16¾	17⅛	17⅝	21	15	18¾	19¾
3	22¾	22	26	24	30	26¼	33¾	33	20¼	20⅜	21⅝	27½	18	22¾	22½
3½	25¾	24½	29½	27¼	34¼	28¾	39¼	37¾	22½	23⅜	24⅞	31⅞	21	25¾	27
4	28⅝	31	31	31	39	30½	41¼	43	22⅞	23½	29	34	24	29⅝	30¼
4½	31¼	34½	34½	34	44	34¼	41¼	49	24¾	25¾	34	38⅞	28	29½	34¼
5	33½	35½	35½	37	49	36⅛	47	55	27	28	39	42⅜	32	33½	38¼
5½	37¼	40½	40½	40½	53½	40½	51	60	29	30¼	42	47	36	37¼	44
6	42½	46	46	44	58	41½	55	65	33	34½	44	52¼	42	42½	51
7	48½	49	49	50	68	45½	64	77	36¼	38½	53	60½	48	48½	57
8	54⅝	57	63	57	77	49⅛	72	87	38	39¾	60	68½	48	48⅛	59
9	60¾	63	73	65	87	57½	80	98	41	42¾	68	77	54	54¾	65
10	72¾	66	84	72	96	68½	88	108	44⅜	47	72	85½	60	60¾	71

TABLE VI. DIMENSIONS OF THREE-QUARTER HOUSING FANS.

Size of Wheel.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P
4	28⅝	29½	36	31	39	30½	15	43	22⅞	23½	29	34	24	24⅝	35
4½	31¼	32	40	34	44	34¼	18	49	24⅞	25¾	34	38⅞	28	29½	39
5	33½	34½	44	37	49	36⅛	20	55	26¾	27¾	39	42⅜	32	33½	43
5½	37¼	38¼	48¾	40½	53½	40½	20	60	29	30¼	42	47	36	37¼	47
6	42½	40¾	52	44	58	41½	24	65	33	34½	44	52¼	42	42½	51
7	48½	47	62	50	68	45½	26¼	77	36¼	38½	53	60½	48	48½	59
8	54½	52½	69	57	77	49½	32¼	87	38	39¾	60	68½	48	48½	67
9	60¾	60	78	65	87	57½	36	98	41	42¾	68	77	54	54¾	76
10	72¾	66	85	72	96	68½	40	108	44⅜	47	72	85½	60	60¾	84

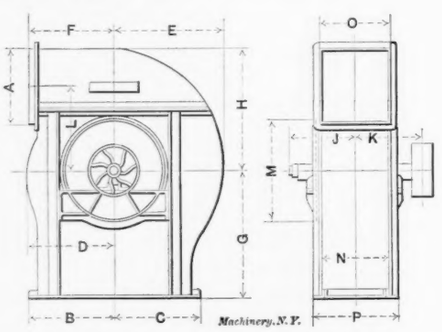


Diagram of Full Housing, use with Table V.

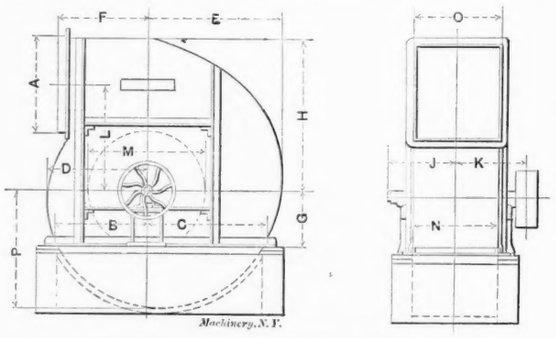


Diagram of Three-quarter Housing, use with Table VI.

Table VII. gives the blast areas for fans of different diameters, computed by the above method, which will be found to correspond very closely with those calculated by more complex methods for fans of approximately the same proportions.

TABLE VII.

Dia. of Fan in feet.	Blast Area, in sq. ft.
3	1.26
3½	1.72
4	2.24
4½	2.84
5	3.50
6	5.04
7	6.86
8	8.96
9	11.34
10	14.00

Actual Capacity.

In the examples given under *Theoretical Capacity*, it was assumed that the effective area of outlet was equal to the blast area, so that the velocity of outflow could be taken the same as the tip velocity. In actual practice the effective area of outlet is always made greater than the blast area and consequently the actual volume of air discharged is greater than the theoretical. On the other hand, the pressure drops below that due to the tip velocity and the velocity of flow through the outlet is correspondingly less.

The size of discharge outlet varies somewhat for different makes, but for a large number of fans examined it was found to average about 2.23 times the blast area as computed by the preceding method. Assuming a coefficient of discharge of .8, it gives as the effective area of discharge, $.8 \times 2.23 = 1.78$ times the blast area.

A series of carefully conducted tests made some time ago upon an enclosed fan of practically the proportions taken, showed the pressure producing the flow of air through the outlet to be about .7 of that due to the peripheral velocity, when the effective area of outlet was made 1.78 times the blast area as computed above. Calculations based upon tests made by one of the leading manufacturers of fans of similar proportions give practically the same result.

We have seen that the velocity corresponding to any given pressure, or in other words, the peripheral velocity necessary to produce any desired pressure may be found by the formula:

$$v = 65.5 \sqrt{h}$$

for air at 50 degrees temperature, when the effective area of outlet is equal to or less than the blast area.

TABLE VIII.

Dia. of Fan.	¼ oz.	⅜ oz.	½ oz.	⅝ oz.	¾ oz.	⅞ oz.	1 oz.	1¼ oz.
3	5,690	6,960	4,380	8,980	9,840	10,600	11,350	12,700
3½	7,750	9,490	10,950	12,250	13,400	14,500	15,500	17,300
4	10,350	12,650	14,600	16,350	17,900	19,300	20,650	23,100
4½	12,950	15,850	18,250	20,400	22,350	24,150	25,800	28,850
5	16,050	19,600	22,650	25,500	27,750	29,950	32,000	35,750
6	23,250	28,600	32,900	36,750	40,250	43,450	46,450	51,900
7	31,550	38,600	44,600	49,800	54,550	58,950	62,950	70,350
8	40,850	50,000	57,750	64,550	70,650	76,300	81,550	91,150
9	53,200	63,950	73,750	82,500	90,350	97,560	104,250	116,500
10	64,100	78,500	90,600	101,300	110,950	119,800	128,000	143,050

If increasing the effective outlet area to 1.78 times the blast area causes the pressure to drop to .7 that due to the peripheral velocity, we must, in order to again bring the pressure up to its original point, increase the velocity of the fan tips to a speed given by the equation

$$v = 65.5 \sqrt{\frac{h}{.7}}$$

Assuming an original pressure (h) of 1, it is found that the tip velocity must be increased by 1.2 in order to produce this result.

This may be made clearer by an illustration: A 6-foot fan running at a speed of 194 revolutions per minute produces a pressure of ½ oz. with a discharge outlet having an effective area equal to the blast area. If the effective discharge outlet is made 1.78 times the blast area, at what speed must the fan be run to maintain the same pressure; that is ½ ounce?

$194 \times 1.2 = 233$ revolutions per minute.

Table VIII. gives the cubic feet of air discharged per minute by fans of different diameters when run at such speeds as will produce the pressures indicated at the head of each column. These results were obtained by assuming the effective area of discharge outlet equal to 1.78 times the blast area, and multiplying this area in square inches by the quantities for the corresponding pressures as given in Table III. (October MACHINERY.) The results are given to the nearest ten, for quantities less than 10,000, and to the nearest fifty for those above 10,000.

Table IX. gives the speeds of fans of different sizes necessary to maintain various pressures over effective discharge areas equal to 1.78 times the blast areas. These results are obtained by multiplying the speeds given in Table II. by 1.2.

TABLE IX.

Dia. of Fan.	¼ oz.	⅜ oz.	½ oz.	⅝ oz	¾ oz.	⅞ oz.	1 oz.	1¼ oz.
3	328	403	465	531	570	615	657	734
3½	282	345	398	446	488	526	562	630
4	247	302	349	390	427	460	493	550
4½	219	268	309	346	379	410	438	489
5	196	242	278	312	342	369	394	440
6	164	201	232	260	285	307	328	367
7	140	172	199	223	243	264	282	314
8	123	151	175	195	213	230	246	276
9	110	134	154	172	189	205	219	244
10	98	121	139	156	170	184	197	220

Horse-power Required.

The power required for moving a given quantity of air under different conditions is given in Table III. This, however, does not include that necessary for overcoming the friction of the fan or the passage of the air through it.

The efficiency of a fan varies with the speed, the size of outlet, and the pressure against which it is working. Under favorable conditions, properly proportioned fans should have an efficiency of about 40 per cent., although they often fall considerably below this. The horse-power given in Table X. for different sized fans is obtained by multiplying the effective area of outlet, in square inches (blast area × 1.78) by the quantities given in column three of Table III., and dividing the result by .4 (the efficiency).

TABLE X.

Dia. of Fan.	H. P. Required for Different Pressures.							
	¼ oz.	⅜ oz.	½ oz.	⅝ oz.	¾ oz.	⅞ oz.	1 oz.	1¼ oz.
3	1.0	1.9	2.8	3.8	5.0	6.4	7.9	10.8
3½	1.5	2.5	3.8	5.4	7.0	8.8	10.8	15.0
4	1.8	3.3	5.0	6.9	9.1	11.5	14.0	19.6
4½	2.3	4.0	6.3	8.8	11.5	14.5	17.8	24.8
5	2.8	5.0	7.8	10.3	14.3	17.9	21.9	30.5
6	3.9	7.3	11.1	15.3	20.4	25.8	35.1	44.0
7	5.3	10.0	15.3	21.3	27.9	35.1	42.3	60.0
8	7.0	13.0	20.0	28.0	36.7	46.4	56.8	81.0
9	8.8	16.1	25.0	34.8	45.6	57.5	70.4	98.5
10	11.0	20.3	31.1	43.5	57.1	72.0	88.0	122.9

* * *

It is easily demonstrated that motion must always be communicated to a body by pushing it; when drawn it is pushed by the hook or other coupling connecting it to the driving body. Reduced to the last analysis, motion is communicated by the impact of particles one against another. This means compression, and, in accordance with the ordinary conception of the term, a diminution of volume; but diminution of volume is not an invariable accompaniment of compression as a result of transmission of motion. An ingenious and simple method of illustrating this fact, and at the same time illustrating the physicists' conception of the effect of motion in the ether, is by the use of a rubber bag filled with shot. In a state of quiescence the shot lie closely together; but if motion is communicated to the shot by pressure directed against any portion of the bag its volume at once increases. This comes about for the simple reason that each individual shot requires more room when it is moving relative to its neighbors than when it is stationary.

A FLOOR PLATE BORING MILL AT THE CROCKER-WHEELER WORKS.

Perhaps the most radical change in the past ten years in the building of machinery is that which has taken place in the tools and methods employed in the construction of heavy electrical apparatus. The method by which this is mainly done nowadays is that known as the floor plate system. It consists, briefly, in fastening the work in its rough state to a suitably constructed floor plate, and performing as many as possible of the operations necessary to finish it without removing it from its fastenings. Various tools, such as drilling machines, slotters, milling machines, etc., are brought to the proper position by traveling cranes, fastened to the floor plate, and set at work; as many of these machines being employed on one casting as the size of the work and the convenience of the operations will allow. The machine which is described in the following paragraphs is a logical development of this system. Perhaps it would be nearer the truth to call it a revolving floor plate rather than a boring mill, as its functions seem to be rather those of the floor plate than of the machine which we commonly think of as the boring mill.

The Crocker-Wheeler Co., Ampere, N. J., have recently gone into the business of making alternating current machinery, and are constructing generators of the largest capacity for this work. To keep pace with the demands for this product, they are building a new shop for handling the heavier part of the work. This building is about 60 feet wide and will eventually be about 500 feet long, although at present only enough of it is being completed to furnish quarters for the machine under discussion. The floor of the central bay will be made up of surface plates. These plates are built in sections 6 x 10 feet and mounted on concrete piers about 7 feet high, spaced far enough apart to allow easy access between them for the purpose of keeping the plates in alignment, and cleaning out the chips and dirt which fall down through the openings in the floor plate. The building is served by a 30-ton Niles traveling crane of 60 feet span.

Fig. 1 shows a vertical section through the center of the machine. It will be seen to consist essentially of a stationary center *A*, resting on cement piers, which acts as a support for a revolving annular platen or face plate. This is flush with the main floor plate of the building in which it is installed, and when not in use is simply a part of the floor of the building. The working surface of this revolving platen is composed of 16 T-slot segments. The platen was given this sectional form to facilitate repairs in case at any future time the T-slot edges should be broken, or the face plate surface itself cracked. The face plate is furnished with a hub *B* to which it is keyed, and which revolves in a suitable bearing on the base casting *A*. A tapered seat in the base casting locates this hub centrally, and adjustment for wear is maintained at this point by means of a sleeve *C* keyed to hub *B* whose outside surface is cylindrical, and whose inside surface is bored to the taper of the main casting. A packing ring *D* is furnished to make a tight joint between the revolving sleeve and the stationary casting, thus preventing dirt and chips from working down into the bearings. The lower surface of hub *B* seats on a horizontal annular bearing surface which is provided with a raised lip to retain the oil used for lubrication.

To a seat on the under side of the periphery of the face plate is bolted a circular cast-iron rack *E* made in eight sections. The teeth of this rack are one diametral pitch, and are cut from the solid. The pitch diameter is 18 feet, and the length of the teeth is 10 inches. A 16-tooth cast steel pinion *F* on a vertical shaft meshes with this circular rack and thus drives the table. The under side of the rack is made in the form of a flange to furnish a "safety bearing" for the table.

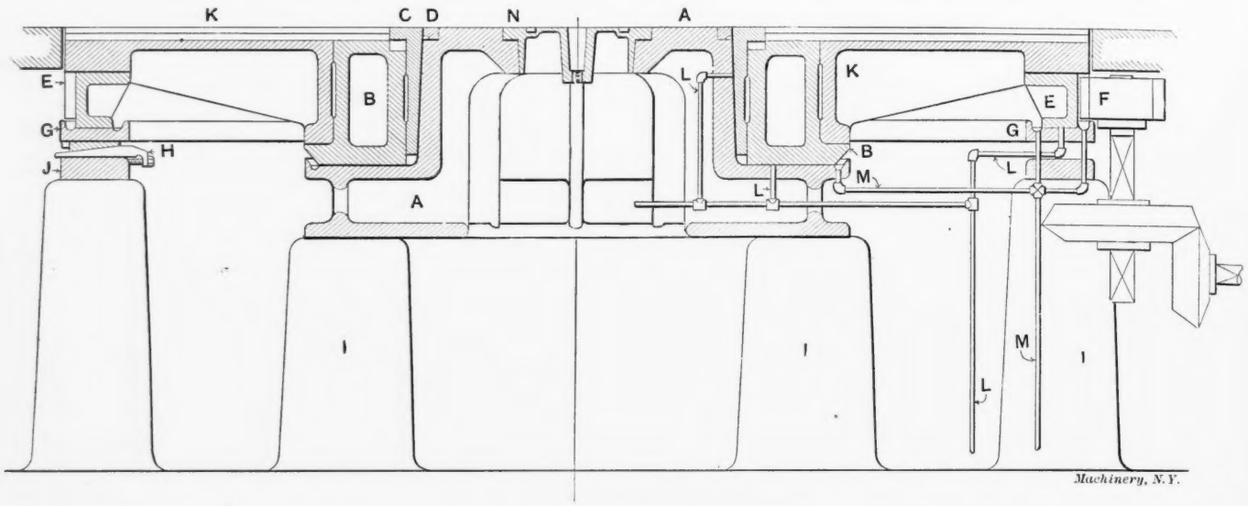


Fig. 1. Cross-section of Boring Mill Table.

This rests on a corresponding bearing surface on ring *G* which is furnished with lips on each side to retain the lubricating oil. This ring is supported by wedges *H* of which there are 32, evenly spaced around the circumference and supported by ring *J*, which seats on concrete piers *I*. In setting up the machine the bearing surfaces on the main casting *A* are first adjusted properly and then ring *G* is raised by means of wedges *H* until it just touches the under surface of the table gear *E*. It is not set up so far that it takes any of the weight of the table under normal conditions. If, however, the table

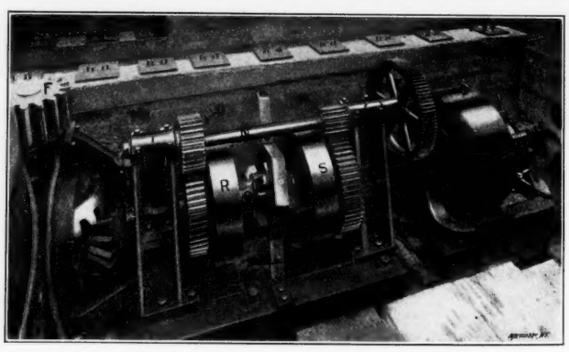


Fig. 3. Driving Mechanism.

be heavily loaded at any time, this bearing will come into play, and take care of any springing or deflection of the table. Fig. 2 is a reproduction of a photograph taken while the machine was being erected, and shows the table just being put into place. It gives a good idea of the different parts of the machine. In Fig. 3 is shown the table driving mechanism. Power is furnished by a 50-horse power standard shunt wound motor, and the speed changes are obtained partly through gearing and partly in the motor by use of the multiple voltage system. The connections in the controller give voltages of 40, 80, 120, 160, and 240 and field regulation provides for intermediate gradations. The motor is geared to the shaft seen at *N* in Fig. 3. On this shaft are mounted two pinions *O* and

P, which mesh with mating gears on the shaft *Q*. Either of these two gears may be connected by clutches *R* and *S* to shaft *Q*. This provides for a speed change ratio of about 2.46 to 1, and with the motor changes drives the table at a rate varying from .34 to 3.8 R. P. M. Shaft *Q* is connected to the vertical shaft which revolves the table through bevel gears at the extreme left hand end. A sheet metal roof covers this driving mechanism to protect it from chips and dirt which fall down through the openings in the floor plate above.

The bearings are lubricated by a forced oiling system, of which supply pipes *L L* and drain pipes *M M* may be seen in Fig. 1. In the vertical portion of the center bearing for instance, a supply pipe is tapped into the bearing at four equidistant points around the circle. Each pipe leads out into a vertical groove extending the height of the bearing. On the inner surface of sleeve *C* there is cut an undulating oil groove. As this sleeve revolves on the bearing, oil from the vertical in the main casting is introduced to this groove in the sleeve at every point along its length. These grooves are thus kept full of oil, which is wiped over every part of the bearing surface in both the rotating and stationary members of the bearing.

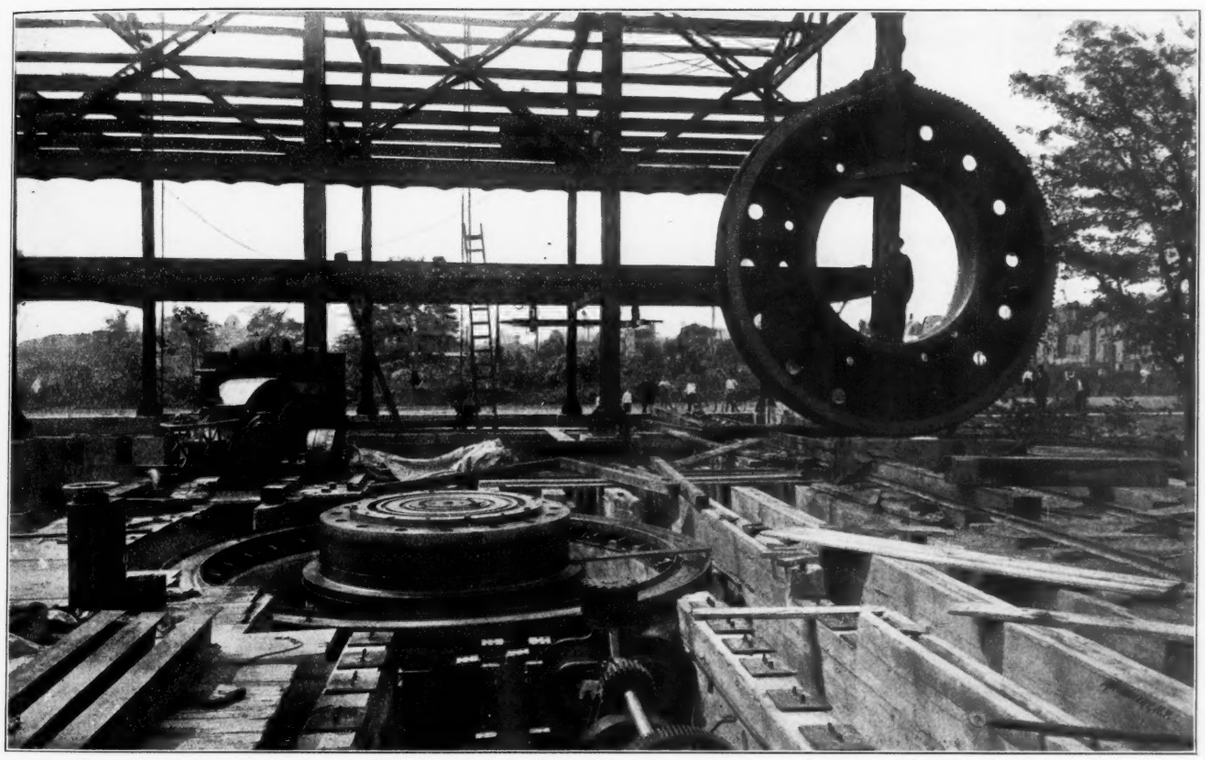


Fig. 2. The Machine in Process of Erection.

Similar provision is made at the lower bearing of the main casting, at the safety bearing on the outer diameter of the table, and at all the bearings in the driving mechanism. After being drained from the outer lips of the bearing surfaces, the oil passes through a filter and is then returned to the reservoir to be used over again indefinitely.

Shortly after the machine was erected, this oiling system was tested; a pump was provided capable of giving 60 pounds pressure to the square inch. Such pressure, however, was found to be entirely unnecessary, as the head of oil in the barrel which furnished the supply reservoir was sufficient to furnish a copious supply of lubricant to all the surfaces. This head amounted to only about two feet, so the pressure was less than one pound to the square inch.

This machine differs from others which have been built for somewhat similar purposes in the construction which gives a stationary center to the revolving platen and in the fact that there is nothing which corresponds to the housings and cross-rails which are usually considered as a necessary part of a boring mill. This machine is designed to use portable tools and housings entirely. The work and tools may thus be set up in a variety of different ways. A tool holder may be clamped to the stationary part *A* in the center to take a cut on the inside diameter of a casting fastened to the revolving table. If the work is too large to admit of this being done, this tool holder may be mounted on a supplementary plate which has been furnished to extend the diameter of this stationary central part. By this means the radius of action of a stationary tool may be increased to bore a hole of about thirteen feet in diameter. For still larger boring the work may be mounted on the floor plate outside of the machine, while the tool post or housing is fastened to the revolving platen near its outside periphery. In working on the outside diameter of castings, the work may be fastened to the stationary center while the tool revolves about it, or on larger work the casting may be mounted on the platen while the tool is clamped to the stationary floor plate. Later on, provision will be made for boring the hubs of rotors and similar parts while the outside surface is being machined. For this purpose a boring bar provided with an individual drive will extend up through the opening in the stationary plate, which is normally filled by gage plug *N*. On this revolving boring bar will slide a head on which the cutters are mounted. Means will be provided for feeding this head up or down the revolving bar. Thus the outside and inside surfaces of the rotors may be finished, each tool working at its proper cutting speed.

The equipment of portable tools at present provided comprises the tool post or housing shown in outline in Fig. 4; the boring, drilling and milling machine which was illustrated among new tools in the October issue of MACHINERY; and a portable slotting machine. The first two tools were built by the Espen-Lucas Co., the latter by the Newton Machine Tool Works, both of Philadelphia, Pa. In the tool post in Fig. 4, the only motions are those necessary for the feed, which may be set to give the tool either a vertical or a horizontal movement. Speeds and feeds are thus seen to be entirely independent of each other. The other two machines, the boring drilling and milling machine and the slotter, are of course self contained units and may be used either in conjunction with this boring mill (which then simply acts as a face plate for holding the work or tools and indexing them to the proper position) or on any other work about the building.

A boring machine or revolving floor plate of this type pos-

sesses certain very evident advantages. Rotors and stators of electrical generators and work of a similar character may be practically completed at one setting by using this machine in combination with portable tools. Again, in work of this character and with tools of this size, it is practically impossible to keep such a machine at work continuously, and there must be long periods when it will be inactive. At such times the platen will simply form a part of the main floor plate of the building, while the portable tools which are ordinarily used in connection with it will be moved away to other parts of the floor, where they may be employed on whatever work needs to be done at the time. The case is thus the same as if one could take the housings and cross-rails from an ordinary boring mill and use them for other purposes when the machine as a whole was not needed, at the same time stowing away the useless parts of the machine so as to leave its place free for whatever work it might be expedient to do there. These considerations suggested to President S. S. Wheeler of the Crocker-Wheeler Co., the main features embodied in the machine.

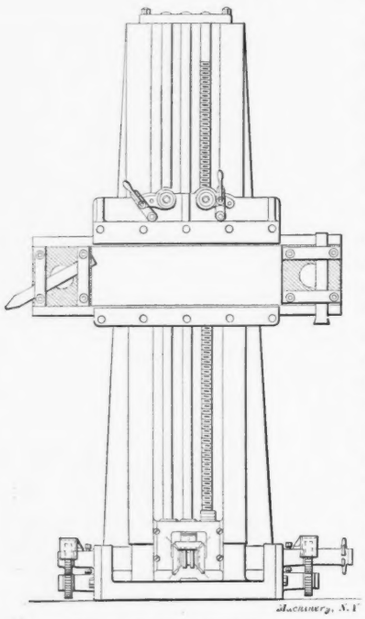


Fig. 4. Portable Tool Post.

Referring to Fig. 1, the weight of base casting *A* is 32,000 pounds; face plate *K* weighs 40,000 pounds; the total weight, exclusive of driving mechanism, is about 90,000 pounds. The face plate is 18 feet in diameter. The stationary center is 5 feet 6 inches in diameter but may be extended to 9 feet, as explained above. The limit in diameter for boring on this machine is about 25 feet; the driving mechanism is so arranged that the surface speed will not be abnormally high for turning the outside of the largest casting which it is possible to easily fasten to the face plate.

A NEW VALVE GEAR EMBODYING A NOVEL MECHANICAL MOVEMENT.

A valve gear has recently been designed by Mr. E. J. Armstrong, of the Ball Engine Co., Erie, Pa., which has some exceptionally interesting features. The gear is now used by the Ball Engine Co. on their four-valve medium-speed engines, in which rotary or Corliss valves are employed. In engines of this type the speed of rotation is sufficient to allow the use of a shaft governor so that the admission and cut-off of the steam can be controlled by a shifting eccentric under the influence of the governor. It is also desirable, at the speeds at which such engines run, to employ a valve gear that will give a positive opening and closing of the valves without the use of the hook cut-off gear and dash pots employed on Corliss engines of the regular type. The gear to be described gives a positive motion to the valves and is so designed that it opens and closes them promptly, but after they are closed allows them to remain in a stationary position until the beginning of the next stroke, when admission occurs.

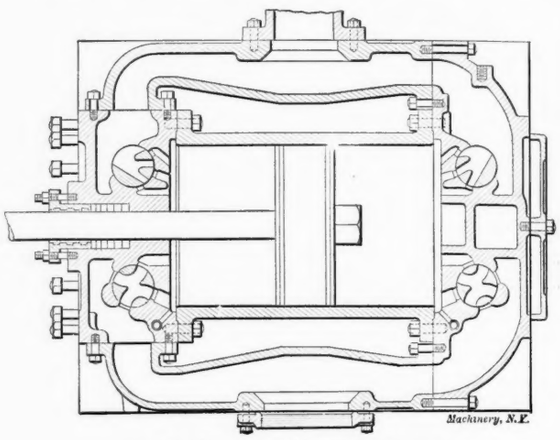


Fig. Section of Cylinder of Ball Engine.

In Fig. 1 is a longitudinal section of the cylinder of this engine. The admission valves are triple-ported, permitting a quick opening and closing, without too rapid movement of the valves. The multi-ported construction also gives a more uniform pressure on the valve seat. The exhaust valves are double ported.

The admission valves of the engine are controlled by a Robb-Armstrong-Sweet shaft governor, while the exhaust valves are operated by a fixed eccentric. The connections with the exhaust valves do not differ essentially from those usually employed on four-valve engines, since the desired conditions of opening and closing can be easily obtained by means of the usual rockers or wrist plate, and links connecting the rockers with the valve levers. The gear for operating the admission valves, and which is controlled by the shifting eccentric is a radical departure, however, and embodies what is probably an entirely new mechanical movement. In most shaft-governed engines of the four-valve type, the admission valves move at all times except with very late cut-offs, and the greatest strain of moving them while unbalanced by the steam pressure comes as the eccentric passes its centers, causing

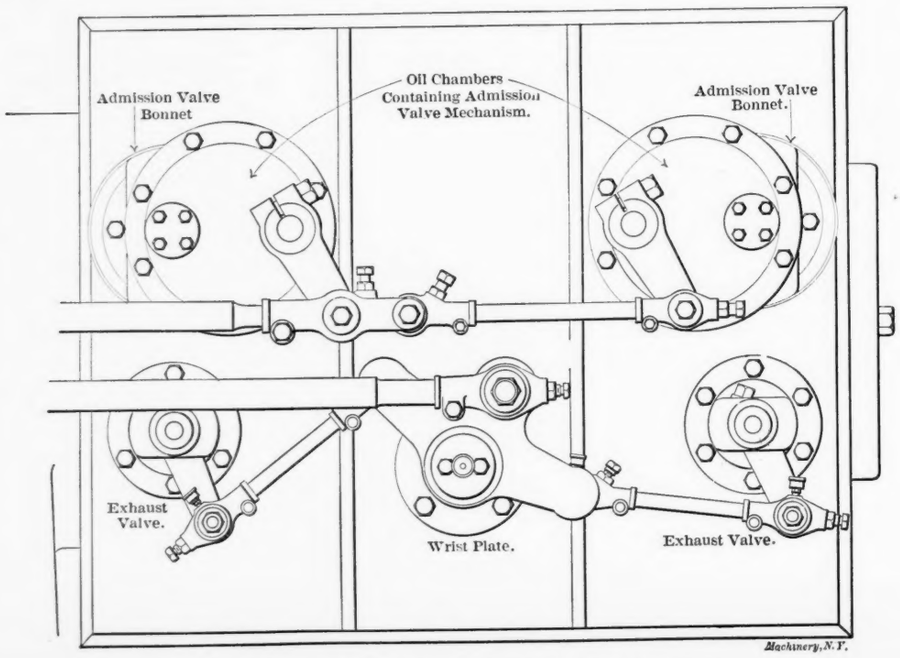


Fig. 2 Diagram showing General Appearance of New Valve Gear.

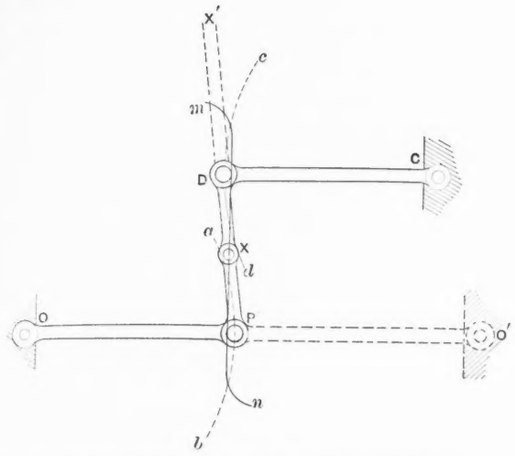


Fig. 3. The Watt Parallel Motion, which is the Basis of the New Valve Gear.

a tendency to pound. As previously stated, however, the new Armstrong gear allows the valve to remain stationary, between the time of closure and the time when it must move to admit steam, which reduces the strain upon the gear. The motion is so arranged, moreover, that when the valve does move to admit steam the eccentric is at the middle of its stroke when all slack has been taken up and so does not produce a pound.

The essential features of the valve motion are embodied in the rockers, levers and their connections upon the cylinder, which take the place of the usual Corliss wrist-plate and connections. The motion for operating the admission valves is a modification of the well-known Watt's parallel or straight-line motion, shown in Fig. 3. It consists of two levers *C D* and *O P*, pivoted at points *C* and *O* and having their outer ends connected by links *D P*. If the levers are of equal length, a point *X* located upon the links midway between points *D* and *P* will move approximately in a straight line, through a certain range of motion. If, however, the links should oscillate through too great a range of motion, point *X* would deviate from a straight line and the path that it would trace throughout its total movement would be line *mn*, Fig. 3. It will be noted that the central portion of this line is straight, while the upper and lower ends are curved, where the deviation from the straight path occurs.

It is possible to produce modifications of this motion, as for example, one lever may be longer than the other, in which case the position of the point upon the link *D P* which will trace a straight line, would be located either above or below the center of the link, according to the proportions of the levers.

A little consideration will show that any point upon the link *D P*, Fig. 3, between point *X* which traces a straight line, and the ends *D* or *P* of the links, will necessarily trace a curved path, being governed in its motion more by the action of one lever than it is by the action of the other. This fact has been taken advantage of in the designing of a valve gear previously used by the Ball Engine Co., and which has operated successfully. A diagram of this gear is shown in Fig. 4. One arm of a rocker corresponding to the wrist plate of a Corliss engine, is represented by the line *C D*. This rocker pivots on point *C* and is connected by link *D P* to a second rocker pivoted at point *O*. The two rockers *C D* and *O P*, and the connecting link *D P* constitute the Watt straight-line motion; only, the point *X* upon the link *D P* is so located that it will trace approximately the arc of a circle *h l* instead of a straight line.

The connection with the valve lever attached to each of the admission valves of the engine is *X F*, point *E* representing the axis of the valve. We will now suppose point *D* at the outer end of the wrist-plate rocker to move downward on the arc *c d*. The other rocker *O P* will consequently move downward, the end *P* tracing a curved path *o p*. Point *X* on the connecting link will follow a curved path during a portion of its travel, which path is the arc of a circle having its center at the point *F* of the valve lever. The valve lever will therefore not oscillate as long as point *X* of the connecting link travels on this arc, as it does through a certain range of motion, in accordance with the same principle that point *X* in Fig. 3 traces a straight line within a certain range of motion. Beyond this range, however, point *X* in Fig. 4 will depart from the arc of the circle having *F* as a center, and the valve lever will then oscillate to the right, opening the valve. This

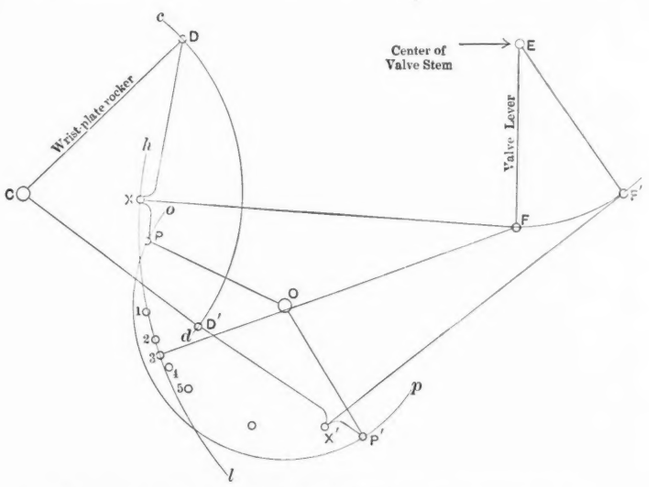


Fig. 4. Diagram showing Layout of Valve Gear as previously made.

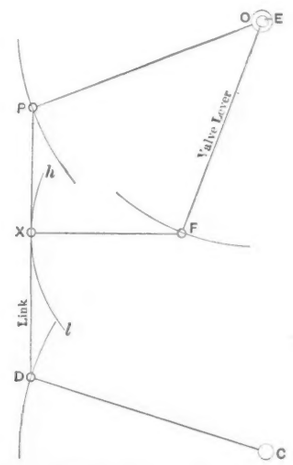


Fig. 5. Diagram of Improved Valve Gear.

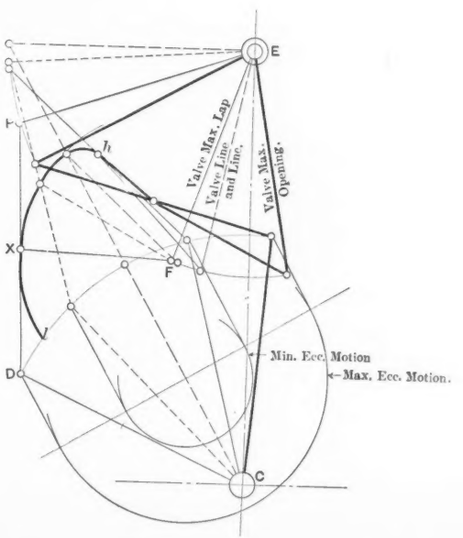


Fig. 6. Layout of Improved Gear in Several Positions.

departure from the arc occurs at point 3. Successive positions of point X are indicated by the small circles 1, 2, 3, 4, etc. When the end D of the wrist-plate rocker has reached its extreme position, it will be at D' and then point P will be at P', point X at X' and point F of the valve lever at point F'.

The new gear that the Ball Engine Co. have just brought out embodies the same principle as the gear just described, which as explained is a modification of Watt's parallel motion. It is, however, arranged in a much more compact form; so compact, in fact, that the various levers and connections are encased in a small cast iron cylinder filled with oil and attached to the bonnet of each admission valve.

The modification of the Watt parallel motion used in the new gear consists in placing both levers on the same side of the connecting link as indicated by the dotted lines in Fig. 3, where the lever is shown in position $O'P$. With such a construction, however, the motion can be used to trace a straight line only by extending the link DP beyond one of the levers so that the point which is to describe the straight line will be located in a position indicated approximately by X' in Fig. 3. Any point as X located upon the link between the two levers will necessarily follow the arc of a circle within certain limits of motion, and then, as before, will depart from this path, following a curve of smaller radius.

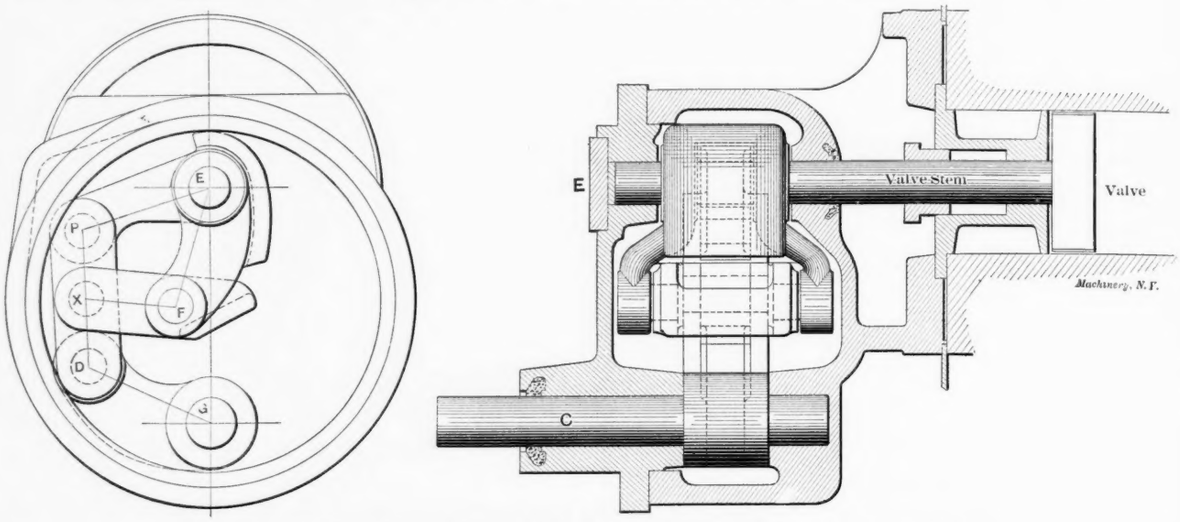


Fig. 7. Arrangement of Levers and Section of Bonnet and Valve Gear Chamber, the latter containing Oil.

Fig. 5 shows in outline the arrangement of this modification as worked out in the new type of gear. The different elements are lettered in Fig. 5 to correspond with the letters of Fig. 4. CD is the rocker which derives its motion from the eccentric rod, and serves the same purpose as the wrist-plate of the Corliss engine. It is connected by the link DP with a second lever, PO, which is pivoted upon and turns loosely upon the valve stem E. The central point X on the link PD describes the arc of a circle when the two levers oscillate back and forth. Point X is connected by the link XF with the valve lever FE which is keyed to the valve stem at E. In this motion, as before, point F is the center of the arc $h\,l$, traced by point X. As before, point X follows this arc $h\,l$ only within a certain range of motion, and when this range is exceeded it begins to describe an arc having a shorter radius, the lever $E\,F$, therefore, begins to move to the right, and steam is admitted. When the mechanism has reached its extreme limit of motion and begins to move in the opposite direction again, the valve lever EF begins to move to the right, closing the valve and then remains stationary during the balance of the movement gear. In Fig. 6 the motion is shown plotted with the levers and their connections in several positions and the path $h\,l$ of the point X is drawn throughout the range of motion so as to indicate the way in which the radius changes at the point where the valve is to open. Fig. 7 is made from a working drawing of the gear and shows the levers which operate one admission valve.

The Robb Engineering Co., Amherst, Nova Scotia, are the Canadian builders of this type of valve gear.

THE DATA SHEET FOR NOVEMBER.

The table of squares of mixed numbers furnished with this issue of MACHINERY may require a little explanation, as regards the use of the column marked "End Figures." Supposing, for instance, we wish to find the square of the fraction 4 17-32. We follow down the column marked 4 until we come to the row containing the values for 17-32. The quantity given here is 20.5322. Following along this row to the last column, we find given the end figures 265625. This is to be suffixed to the previous figures, giving as a result the quantity 20.5322265625 as the exact result for the square of 4 17-32. In the same way the square of 11 23-64 is 129.035400390625. Again, the square of 2 1-16 is 4.25390625. These figures are carried out to an exact result.

Mr. G. R. Carothers, of Philadelphia, Pa., who computed the table, has found it especially valuable in making the calculations necessary for locating bushings in jig making. The problem of finding the hypothenuse of the right angle triangle when the length of the two sizes is known is of constant occurrence. The use of the table appreciably reduces the labor of finding the hypothenuse. For instance, supposing one side of the triangle is 2 3-8 while the other side is 4 13-16. The operation of obtaining the hypothenuse is as follows:

$$2\tfrac{3}{8}^2 = 5.640625$$
$$4\tfrac{13}{16}^2 = 23.16015625$$
$$28.80078125$$
$$\sqrt{28,80078125} = 5.3666 +$$

The mathematical work is thus reduced to one addition and the extraction of one square root.

* * *

The growth in favor of concrete for chimney construction is indicated by the fact that a little over one year ago the highest concrete-steel chimney was that built for the Pacific Electric Co., at Los Angeles, Cal. It has a height of 158.5 feet above the ground level and 174 feet above the foundation, and has an inside diameter of 11 feet. Since then the Portland General Electric Co. has built a concrete-steel chimney 230 feet high, having a flue 12 feet in diameter. Another built for the Tacoma Smelting Co. in Tacoma, Washington, is 300 feet high and has a flue diameter of 18 feet. But this latter will soon be eclipsed by one in erection at Butte, Montana, which will be 450 feet high when completed. The concrete-steel has a number of advantages for chimney construction, one of which is that it presents a smooth surface both on the exterior and interior; another, that it is a good non-conductor of heat, and still another, is that it requires no lining, being able to resist any temperature up to 1,500 degrees F., according to the statement made by Mr. George C. Mason in an article on the design and construction of tall chimneys of reinforced concrete, which appeared in the October issue of the *Journal of Electricity, Power and Gas.*

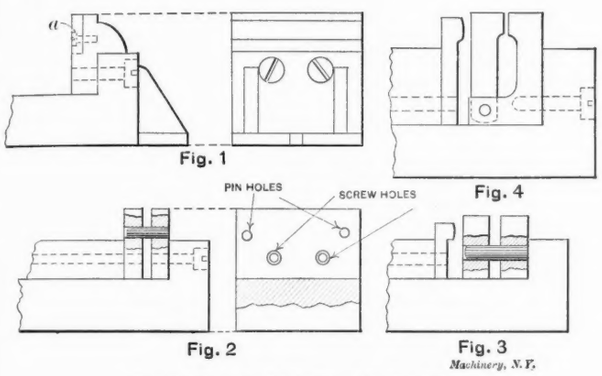
MILLING MACHINE FIXTURES.—1.

E. R. MARKHAM.

The principal consideration, when designing fixtures that are to be fastened solidly to the table of a milling machine, should be to have the table firm enough to admit working the machine and cutter to their limit. In fact, the fixture should be stronger than the machine itself, and able to resist any possible strain that the cutter can exert.

While fixtures should be strong, the movable parts should be so made as to be easily manipulated. All bearing and locating points should be accessible to facilitate the removal of chips and dirt. These not only make the work out of true and small for gage, but chips are pressed into the surfaces, causing chip marks, which every workman despises.

The first fixture to consider is the milling machine vise, which has a stationary and a movable jaw, against which are placed removable jaws, held in place by means of screws. The stationary-removable jaw generally has connected with it any shelf, pins, or means of locating the pieces to be machined. The reason for attaching them to this jaw is that this portion of the vise is not movable, and is, or should be, stiff enough to resist without springing any pressure that may be exerted by means of the crank and screw. The jaw attached to the movable slide part of the vise, on the contrary, is liable to alter its location slightly under strain, especially when the vise becomes worn.



Special Jaws for the Milling Machine Vise.

For some purposes, where but a few pieces are to be milled, or where the character of the pieces is such that there is not much liability of the jaws wearing, and thus affecting the accuracy of the pieces, it is safe to make the jaws of cast iron. If, however, there is a considerable strain on the jaws, it is advisable to make them of steel and harden them. For most purposes, jaws made of a good grade of machinery steel and properly case-hardened answer as well as those made from tool steel, and cost only a fraction as much for stock.

If possible, the piece to be machined should be held in the jaws below the level of the top of the vise in order to avoid springing the jaws out of a vertical position, as would be the case if the piece were above the level of the top of the vise. Occasionally pieces are so shaped, however, that they have to project considerably above the top of the vise jaws, in which case the jaws may be made with a rib which extends over the top of the vise and rests on the piece, as shown in Fig. 1. This furnishes a brace and prevents the springing that would prove harmful to almost any piece of work that it would be safe to hold in a vise while milling.

As it would prove quite expensive if many jaws of this style were made from steel, they may be made from cast iron, and a plate of steel placed where the work is to rest, as shown at *a*, Fig. 1. After the steel plate has been cut to shape and the locating device attached, the jaw may be hardened. If the devices mentioned are pieces which must be attached to the jaw, or pins which enter holes in it, they must be removed when the jaw is hardened. They may be hardened separately and attached afterward, as it would not do to harden two pieces which were screwed together or to harden a jaw with a pin in it.

At times it is necessary to hold pieces so that they rest on shelves on each jaw or are located by pins in both stationary and movable jaw. Generally speaking, it is advisable to construct special fixtures for such pieces, provided the degree of accuracy and the number of pieces warrant the outlay. However, if the pieces must be held in jaws in the vise, some method should be found to prevent the movable jaw from rising when pressure is applied, in the operation of "tightening up."

If the jaws are reasonably thick, large pins may be used, one near each end of the jaw, as shown in Fig. 2. These pins must be forced solidly into one jaw and fit closely in the other. Another method which works nicely is shown in Fig. 3. In this case the movable jaw proper is connected with the stationary jaw by means of pins, or a slide of different design. It is not, however, attached to the movable slide of the vise, but a hardened piece of steel is attached to this and bears against the movable part of the jaw. Many other forms are made, one of which is shown in Fig. 4. The front portion hinges at the bottom, and is pressed against the work by a movable slide. In all such holding devices, however, chips are liable to get between the various parts, decreasing their efficiency.

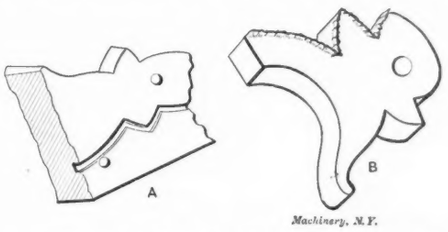


Fig. 5. Arrangement of Fixture or Vise Jaw to Accommodate Burr.

When making any form of holding device, it is necessary to provide a place for the burrs that are a result of previous operations, unless they are removed by a process of filing or grinding. In many cases these burrs will be removed by future operations if it is possible to provide a place for them so they will in no way affect the accuracy of the piece. For this reason milling machine jaws and other fixtures are cut away in places to allow the burrs a place to go, as shown in Fig. 5 at *A*. At *B* a piece of work is shown with the burr mentioned.

It is the custom in most shops to provide a liberal supply of oil, or other lubricant, for cutters when milling work that requires lubrication. In many cases this fluid is used to wash out the jaws or fixtures after removing a piece of work. As this supply is used over and over, however, it is liable to become thick and gummy, and apt to prove harmful rather than helpful, unless the operator watches his fixtures closely. In some shops compressed air is used to blow chips from the working surface, and in many cases "works like a charm." On certain jobs nothing seems so effective as the hand and finger method for cleaning the surfaces of the fixtures.

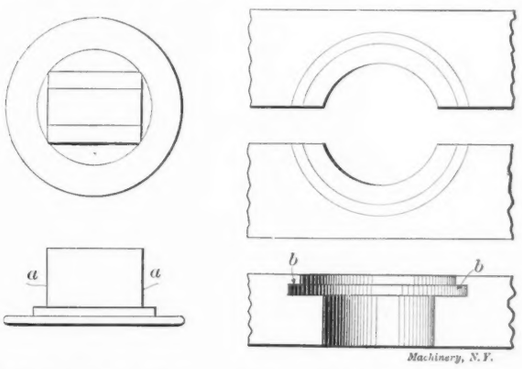


Fig. 6. A Difficult Piece of Straddle Milling.

Fig. 7. Arrangement of Vise Jaws to hold Piece shown in Fig. 6.

At times it is necessary to change existing methods in order to accomplish the desired result. The writer recalls a piece of work consisting of a flange, as shown in Fig. 6, having projecting portions, *a a*, which were to be straddle-milled. The jaws of the vise used to hold this piece had circular grooves,

b b, Fig. 7, which were thought necessary to properly hold the piece, since the pull of the cutters was in an upward direction; but these grooves made an excellent place for a deposit of chips, and as it was a difficult matter to clean them, and as the operator was working by the piece at a rather low rate, the edges of the flanges of the piece being milled became badly scored, and required an extra operation in the turret lathe to remove the marks. To overcome this difficulty, the projecting lips of the vise jaws were cut away and the direction of rotation of the cutters reversed, the overhead belt being changed so that the cutters would run onto the work, thus holding the work securely down on the seating surface of the jaws.

It sometimes happens that the opening in the vise is not sufficient to take in a long piece of work, in which case the jaws may be made of a form shown in Fig. 8. At other times the vise may be used with the movable jaw of the original form, and with the stationary jaw arranged as in Fig. 9. In this case a flat piece of steel is attached to the outside of the

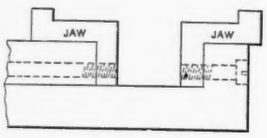


Fig. 8. Offset Vise Jaws.

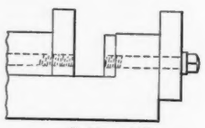


Fig. 9. Extended Vise Jaws.

jaw by means of screws which are a snug fit in holes drilled and reamed in both the auxiliary and stationary jaws of the vise. It is apparent that such an arrangement does not allow great accuracy, as the jaw on the end has no backing, and consequently will easily spring, yet there are instances where it answers the purpose as well as a costly fixture.

If milling machine vises are drilled for screws that hold jaws in such a manner that the jaws will readily go on any vise, much valuable time may be saved. If we are equipping a shop with new machines, this may be readily accomplished, as we may order vises drilled alike and corresponding with some vise already in use, and to which a number of pairs of jaws are fitted.

At one time when given charge of this class of work I looked over my vises and found that but few were drilled so the jaws would interchange. To avoid waiting, I plugged the screw holes in the vises that were different from several that did allow of interchange and drilled them so any jaw that fitted one would fit the other. Of course it was necessary to plug the holes in the jaws and redrill and tap them also. But I was amply repaid for my trouble. Of course I found that the spindles and tables of the machines did not all compare and as a consequence it was necessary many times to slightly alter a guide pin or a shelf, but this did not amount to anything compared with the disadvantage of a long wait when two or more jobs were wanted "right away" and then finding that all must be done on the same machine.

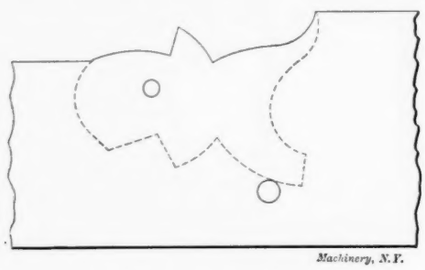


Fig. 10. An Outlined Vise Jaw.

The vises ordinarily furnished with milling machines are opened and closed by means of a screw. Unless it is necessary to apply considerable pressure to the piece being held, this form of vise will not work as quickly as desirable where cheapness of production is a factor. To overcome this objection, vises are made so the slide may be opened and closed by means of a cam and lever, and unless there is much variation in the size of pieces being machined the cam will cause the work to be held sufficiently firm. The work may be placed in and taken out in this way much more quickly than when a vise operated by a screw is used. In fact, where such a vise will answer the purpose, it will be found as cheap to operate and as satisfactory in results as special fixtures; and the jaws necessary when starting a new job are, as a rule, much cheaper than special fixtures.

When it is necessary to cut in the vise jaws the shape of the piece to be milled, it may be done by passing the mills to be used through the jaws as shown in Fig. 10. The pins, or other appliances for holding the work, should now be added, after which they may be removed and the jaws hardened.

Ordinarily such parts of tools as milling machine jaws are made from machinery steel, but I prefer open-hearth steel, which does not contain over 25 or 30 points carbon. This may be casehardened nicely in oil with little or no liability of springing, as the depth of hardness necessary does not call for extreme heat, which causes the steel to go out of shape and also opens the grain of the steel and renders it more liable to become indented should a chip be pressed against the surface.

The jaws may be packed in the hardening box with a mixture of charred bone and wood charcoal—equal quantities—and run five or six hours after they are red hot. Then they may be removed and dipped in a bath of oil, working them up and down lengthwise in the oil until the red has entirely disappeared, after which they may be lowered to the bottom of the tank and allowed to remain until cold.

If for any reason it is necessary to harden the piece deeper than can be done in the length of time mentioned above, then the red-hot jaws may be exposed to the action of the carbonaceous material for a greater length of time. If the jaws are made from the grade of stock mentioned, and are given a low heat, there should be no springing during the hardening process.

* * *

THE PACKING OF MACHINE TOOLS FOR SHIPMENT.

T. S. BENTLEY.

Machine tools and similar articles, which are of considerable value and liable to injury if carelessly handled, require specially careful packing to insure their delivery in perfect condition; yet adequate precautions are often omitted. The faith which some manufacturers appear to repose in the gentleness of those through whose hands their productions have to pass, would be really touching if it were not so obviously misplaced. Many of them seem to suppose that if they stencil the legend "This side up" on the case, they need have no anxiety about the contents shifting, and may neglect all precautions with impunity. Others think it unnecessary even to adopt the warning notice, yet dispatch the machine packed in such a manner that it is almost certain to be badly damaged if the case should be turned over or laid on its side.

Those who have the misfortune to receive the goods at the end of a long journey by rail and sea, are in constant trouble through this inexplicable want of foresight on the part of the makers. Cases only too frequently come to hand bearing unmistakable evidence of having been turned over, dropped, or swung violently against some solid obstruction when suspended. Accidents and rough handling occur with sufficient frequency to be ranked among the contingencies which must be expected and guarded against by all possible means; while the most skillful packing will not always prevent damage, it will often do so, and in any case is certain to minimize risk of breakage. As a matter of fact, however, many machines on their arrival are found to be broken when there are no signs of rough treatment or accident, and the fault is clearly on the part of those responsible for the packing.

Now it generally happens that the damaged machine is wanted for immediate use, and the delay is equally vexatious to the purchaser, whether he can rectify the injury himself, or is forced to await the arrival of replaced parts from the makers. If the trouble is manifestly the fault of the transport companies, the purchaser, however vexed he may be, can make no complaint against the makers; and has usually no means of knowing who is really responsible, or at what part of the

journey the damage occurred. When the packing is clearly to blame, however, his annoyance is focussed upon a definite object; and the matter becomes a serious one for the maker of the machine, for he is not only expected to make good the injured parts, but in addition, his prospects of further business will be prejudiced by the remembrance of the inconvenience or loss occasioned through the negligence of those connected with him.

Merchants and factors who import for re-sale are often put in a very awkward position by reason of occurrences of this kind, as they are dependent on the punctual arrival of the machines in perfect condition to enable them to fulfil their own undertakings with regard to delivery to their customers. Though they cannot prevent delays arising in this manner, they are apt to be held responsible by their clients, and knowing that, they naturally resent being put in a false position through another's fault.

Another thing which is most annoying is to find when a tool is unpacked, that the bright parts are badly rusted, and the appearance of the machine thereby hopelessly spoiled. This may happen through lack of care in applying the slushing oil; in a defective preparation being employed; or through the packing materials being damp when used. Paper which is not thoroughly dry may easily cause trouble in this way.

Some makers seem to imagine that the parts of a machine will travel safely if merely put into a box with some woodwool. While this is a very useful material for packing light pieces, it requires to be stuffed in very tightly, and for heavy parts is quite inadequate. The same remarks apply to sawdust, but in an even greater degree.

While it is impossible to give any hard and fast rules as to the best methods of packing, there are a few points which cannot be too carefully borne in mind. Among them are the following:

1. The case should be made of sound wood, tongued and grooved, and of thickness proportioned to the weight of the machine; when in doubt as to what is necessary, err on the side of strength.

2. It is not sufficient to secure a machine by bolting it down to the bottom of the case. This is often done, and broken bases not infrequently result. The machine should be stayed rigidly in every direction by struts of ample thickness bearing against the solid frame of the machine. Where they bear against the sides of the case, they should not abut against one board only, but should have another piece interposed, which bears crosswire and distributes the pressure over a considerable area. If this method of staying is properly carried out, it effectually secures the machine from moving, and obviates dangerous local stresses. In fact, the best packers rely entirely on this system and use no bolts whatever.

3. The bright parts should be well coated with some good brand of "slushing oil," and special care should be taken that every portion needing protection is properly covered. This is most important, and does not always receive the attention that it deserves; as it is very easy to overlook surfaces which happen to be underneath, or otherwise out of the line of sight, but which require protection all the same. A reliable slushing oil will in most cases be found preferable to either vaseline or the old-fashioned mixture of white lead and tallow. The latter is still used to a considerable extent, and even specially called for by some shipping specifications, but is open to several objections. In the first place, it is very difficult to remove, which causes a good deal of inconvenience to those who have to erect the machines. It is also very unreliable as a rust preventer, and is often found to spoil the appearance of the tool by discoloring the metal upon which it has been spread.

4. For journeys of ordinary length—such as across the Atlantic—no special precautions are usually necessary to guard against damp if the hints named above have been carried out. In wet seasons, however, trouble may be caused through the case being exposed to heavy rain prior to shipment. In some instances this danger is provided against by cautious manufacturers who cover the top of the case with thick waterproof paper, which acts as a roof and keeps the rain from penetrating. This simple expedient is not an expensive matter, and though not generally required, on occasion proves a valuable safeguard.

5. When a machine is taken apart and packed in pieces, it is important that each part be well secured so that it cannot possibly shake loose. The more delicate portions are best packed by themselves in a separate inner case, so that even if anything should get adrift no serious damage could be done.

There is one other matter worthy of passing notice. A very good plan which is adopted by certain makers to lessen the risk of their machines being dropped or turned the wrong way up, is to provide special facilities for slinging the cases in the desired position. As it is natural for everyone to follow the "line of least resistance," it is safe to assume that those who have to handle the cases will do so in the easiest way, and this fact is utilized in the manner indicated.

If the above points were always put into practice, there would be a vast diminution in the amount of trouble experienced through machines arriving in more or less damaged condition; much mutual recrimination would be prevented; and the gain to all parties concerned would be great.

* * *

TANTALUM.

Tantalum is a rare metal discovered by Ekeberg in 1802. It is one of the hardest elements known, and combines with extreme hardness great toughness and ductility. Tantalum filaments have been used with some success in incandescent lamps, and it would not be surprising if its physical qualities should make it useful in other directions. Recent tests made in the Siemens & Halske Works, as reported in the *Zeitschrift für Elektrochemie*, are interesting. A specially pure lump of tantalum was prepared and tested in the department of the works devoted to the manufacture of boring tools, with the following results:

1. With a file of specially hard and fine steel it could be filed, but the file itself suffered in the work.

2. A small chisel of specially hard tool steel, with a point 0.16 inch in breadth, was blunted at every blow of the hammer.

3. A water hardened steel tool marked the mass of tantalum, but the tool speedily lost its edge, and was unable to make any further impression upon the metal.

4. Two boring tools were made from fine grained cast steel, and hardened in quicksilver. One of these tools was of the usual section, the other was triangular. With the first of these tools the thin sheet of tantalum only 0.04 inch in thickness required 7 minutes for perforation, while with the second tool 10 minutes were required, turpentine being used in each case as lubricant for the cutting edge. In each case the cutting edge of the tool was blunted.

From these test results it follows that tantalum, containing only small amounts of impurity, hammered into sheets at a red heat, is equal in hardness to the best and most carefully finished steel. Tantalum, however, greatly exceeds this hard steel in toughness, for while these hard tool steels are brittle, tantalum can be rolled into sheets without injury.

* * *

The origin of most sayings is unknown, or is so obscured as to be very doubtful, especially where it has a local significance. A case in point is that of a well-known saying: "He will never set the North River afire." It is said to have originated in the days of Captain Payne, a freak inventor of the fifties. Captain Payne alleged that he had discovered a process of making illuminating gas from water by electrolysis so cheaply that the ordinary processes would be entirely overthrown. As it afterward turned out the scheme was a fake, but during the height of the excitement, when gas stocks fell to an unprecedented price, the newspapers predicted that water would become the common source of light and fuel, and that New York's heat and light would be drawn from the North or Hudson River. In other words, Captain Payne was thought able to set the North River afire—but he wasn't, hence the quotation originated with the doubters. But the accuracy of this may be doubted, for the British have a similar saying regarding the Thames River.

THE BURSTING STRENGTH OF CAST IRON CYLINDERS.

C. H. BENJAMIN.

Some years ago the writer reported to the American Society of Mechanical Engineers some experiments on the subject indicated by the title of this article.*

In some respects these experiments were unsatisfactory, as many of the cylinders failed at the flanges instead of splitting longitudinally. An attempt to remedy this by thickening the flanges only resulted in the formation of "hot spots" at the junction of shell and flange, with no material increase of strength.

In the same paper was developed a formula for the thickness of such cylinders which assumed the following form:

$$t = \frac{p\,d}{4\,S} + \sqrt{\frac{c\,p\,d^2}{S} + \frac{p^2\,d^2}{16\,S^2}}$$

where

t = thickness of shell in inches,
d = internal diameter in inches,
p = internal pressure in pounds per square inch.
S = tensile strength of metal in pounds per square inch.

The first term under the radical is in the nature of an allowance for bending or distortion of the shell from some lack of uniformity in thickness or in strength, the constant, c, being determined by experiment.

If $c = o$, the equation reduces to

$$t = \frac{p\,d}{2\,S}$$

the usual formula for thin shells.

An examination of several engine cylinders of different makes has shown values of c varying from .03 to .10 with an average value of .06. Experiments on nine different cylinders varying in diameter from 6 to 12 inches gave fairly uniform values for c with an average of

$$c = .05.$$

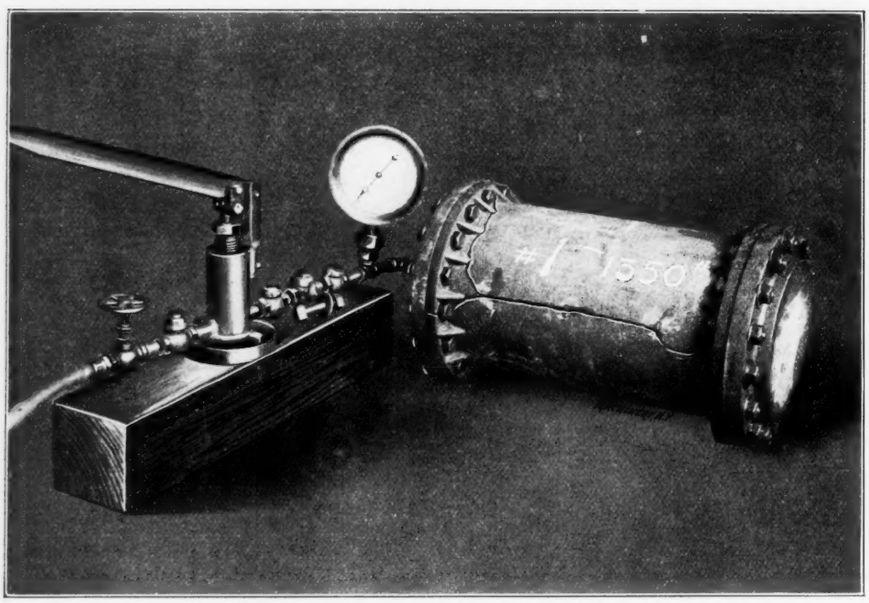


Fig. 1. Apparatus for Testing Cast Iron Cylinders.

The metal of the cylinders was an unusually close-grained, tough cast-iron, having a tensile strength of 24,000 pounds per square inch. The tensile stress on the shell as calculated by the formula:

$$S = \frac{p\,d}{2\,t}$$

averaged about one-third of this, showing the inapplicability of such a formula to cast-iron shells.

In 1903-04 Messrs. A. H. Austin and R. A. Brown made another series of experiments of this character in the laboratories of the Case School, which throws additional light on the problem.

* Trans. A. S. M. E., Vol. XIX.

The experimenters used practically the same apparatus as that described in the paper before referred to. Fig. 1 shows the force pump, the water pipe and check valves, the pressure gage and the cylinder in position for a test. Four cylinders were tested to rupture with water pressure, each cylinder having a length of 20 inches, an internal diameter of 10⅛ inches and a thickness of shell of ¾ inch.

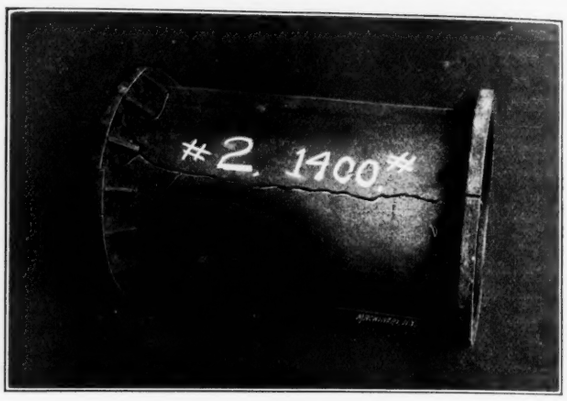


Fig. 2.

The flanges were of the same thickness as the shell and were reinforced by sixteen triangular brackets, as may be seen in Fig. 1. The covers were held to the flanges by sixteen soft steel bolts ⅝-inch in diameter, having a tensile strength of 80,000 pounds per square inch. The heads of the bolts were cut off on one side so as to bring the bolt holes close to the shell and avoid as much as possible the bending moment on the flanges.

Gaskets of straw board soaked in linseed oil and inserted in shallow counterbores were used to prevent leakage. The inside surface of the shell was coated with paraffin for the same reason. Former experiments had shown that water under high pressure would find its way through very minute orifices.

The cylinder heads first used were 1⅛ inch thick and reinforced by 8 radial ribs on the outside. These proved unsatisfactory, the first head breaking at 650 pounds per square inch and the second at 850 pounds. The ribs being on the outside were put in tension by the buckling of the head and had no value. As it was impracticable to put ribs on the inside, the head was thickened to 2⅛ inches at the center, as shown in Fig. 1, when no more trouble was experienced. The four accompanying illustrations show the four cylinders after rupture and the pressure per square inch at the instant of breaking.

It will be noticed that the fractures are all longitudinal, there being but little of the tearing around the shell under the flange, which was so marked a feature of the other experiments. It is evident that the brackets have served the purpose for which they were made in a way that no mere thickening of the flange could do. The metal used for these cylinders was a soft, gray cast-iron having a rather low tensile strength. In fact, pieces cut from the shell of the cylinders and broken in the testing machine had an average value for S of only 14,000 pounds.

As tension specimens of cast iron usually show less than the real strength on account of bending the actual strength of the iron may be slightly more than this.

The average cross-breaking strength of samples from the shell was only 30,000, which is also low.

The following table shows in detail the dimensions and test pressures of the various cylinders. As before stated, the cylinders were all of the same diameter and length, 10⅛ by 20 inches.

BURSTING PRESSURE OF CAST IRON CYLINDERS.

No.	Thickness.			Bursting Pressure.	Value of c.	$S = \frac{p\,d}{2\,t}$
	Max.	Min.	Ave.			
1	.775	.757	.766	1350	.0213	9040
2	.783	.697	.740	1400	.0152	10200
3	.740	.703	.721	1350	.0126	9735
4	.770	.670	.720	1200	.0177	9080

Average value of c = .0167.

The average value of c is thus shown to be only one-third of what it was in cylinders with unsupported flanges. The values for S in the last column give the stress as calculated

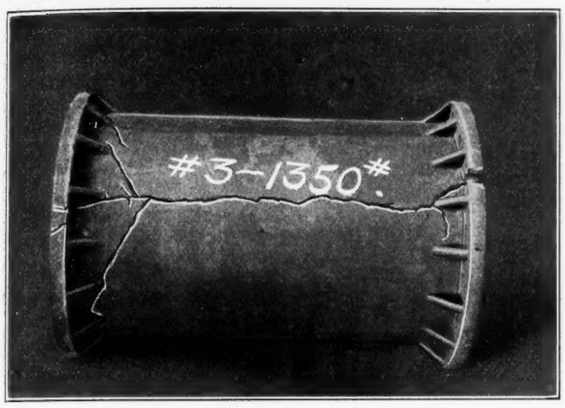


Fig. 3.

by the formula for thin shells, and show that the stress due to bending cannot be neglected even with the reinforced flanges. This may be more clearly shown by solving for S in the formula given at the beginning of the paper:

$$S = \frac{p\,d}{2\,t} + \frac{cpd^2}{t^2}$$

where the first term of the second member gives the stress due to direct tension and the second term the stress due to

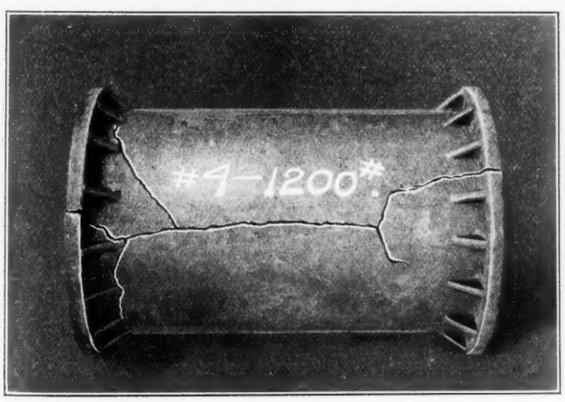


Fig. 4.

bending. Assuming the average value $S = 14{,}000$ as determined by the testing machine, we have for the four cylinders:

No.	S	$\frac{p\,d}{2\,t}$	$\frac{c\,p\,d^2}{t^2}$
1	14,000	9,040	4,960
2	14,000	10,200	3,800
3	14,000	9,735	4,265
4	14,000	9,080	4,920

The "accidental" stress, as it may be called, is seen to be about one-third of the whole.

The fractures in all of the cylinders were longitudinal, beginning at some weak spot near the center and extending either way, usually branching to two or more bolt holes at the flanges.

In this connection it may be of interest to notice a test recently made in the Case laboratories of a gasoline engine cylinder for a Peerless motor car. This cylinder broke around a circumference just above the lower flange when subjected to a hydraulic pressure of 1,600 pounds per square inch. The cylinder had an internal diameter of 4.25 inches and a shell thickness of 5-16 inch. The flange was 9-16 inch thick. The fracture showed a clean, close-grained iron. Assuming a tensile strength of 18,000 pounds per square inch and substituting values, we have $c = .024$.

The conclusions to be derived from these experiments are:

First, that when the cylinder flanges are unsupported, the initial fracture will be circumferential and close to the flange at a pressure very much less than that determined by the formula:

$$p = \frac{2\,t\,S}{d}$$

Second, that when the flanges are sufficiently braced to insure longitudinal fracture, a considerable allowance must be made for bending and other accidental stresses.

* * *

HYDRAULIC CLAMPING CYLINDERS.

FRANK B. KLEINHANS.

Considerable difficulty is experienced in large flanging machines in connection with the clamping cylinders. There are four of these around the main cylinder, each with a certain radial adjustment to suit the various dies with which they are used. The difficulty lies in the fact that these cylinders are connected to one valve, and when water is admitted to them the different plungers advance at different rates of

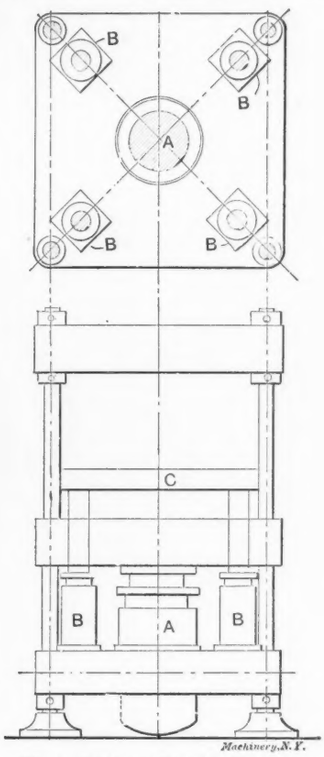


Fig. 1. Sketch of Flanging Machine, showing Main and Auxiliary Cylinders.

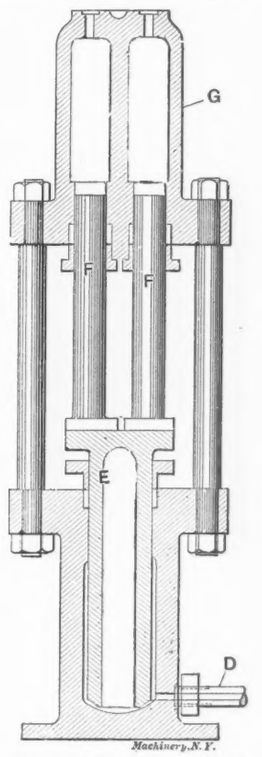


Fig. 2. Hydraulic Equalizer, for Operating the Auxiliary Cylinders in Unison.

speed. Under these circumstances it is impossible to raise the die in a level position and this makes it difficult to do certain classes of work. If it were possible to move all four of these plungers in absolute unison many operations could be performed at a much less cost than at present. Fig. 1 shows an elevation and plan view of the arrangement of the cylinders in a flanging machine. *A* is the main flanging cylinder and the four marked *B* are the auxiliary ones which operate the clamping die *C*. It is the operation of these last that causes the trouble.

An arrangement designed to raise these plungers together

is shown in Fig. 2. The pipe *D* leads from the operating valve to the lower cylinder of the equalizer. The pressure acts upon the plunger *E* and forces it up against four plungers *F*, the combined areas of which are a little less than that of *E*, the reduction being made to offset the weight of the plungers and the slight friction of the stuffing boxes. The top cylinder *G* has four chambers, each of which is independent of the others. Fig. 3 shows the connections between these chambers and the clamping cylinders of the press, the reference letters being the same for all the cuts. *H* is a valve chest at the top of cylinder *G*; *J* and *K* are relief and check valves respectively, and *L* is the supply pipe which replaces water lost by leakage from the system.

In the position shown, the plunger *E* is at the bottom of its travel and the four chambers in cylinder *G* are full of water, as are also pipes leading from them to the adjustable plungers *B*, which are at the bottom of their stroke. If now, water be admitted to the lower equalizer cylinder, the four plungers *F* are moved up equal distances, and force equal amounts of water out of their chambers through the piping to the clamping cylinders. The four plungers *B* must then move in synchronism even though one of them be supporting its full load while the others have no load at all.

There is one difficulty, however, which has to be avoided in this device. It is possible to obtain a dangerously intensified pressure. If all the load be concentrated over one of the clamping plungers, its corresponding chamber in the equalizer will have to take the full fourfold thrust of plunger *E*. To avoid this each chamber is supplied with a relief valve which

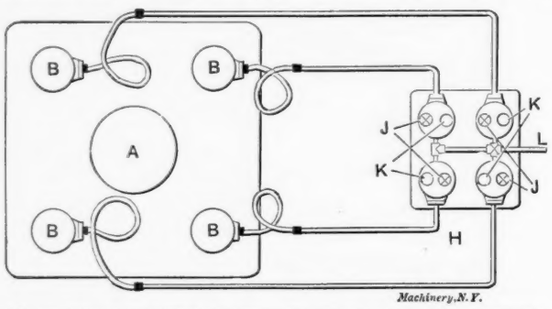


Fig. 3. Diagram of Piping between Equalizer and Flanging Machine.

rises from its seat when the pressure in the chamber exceeds that of the accumulator. Each chamber is supplied in addition with a check valve *K* which makes good any loss of water in the equalizer system by admitting a new supply from pipe *L* when the pressure is released in the chambers. As soon as the operating valve is open again, this valve is closed by the pressure of the water behind it.

* * *

It is somewhat impressive to learn that the gyratory motion imparted to a rifle bullet by the twist in the barrel may reach a peripheral velocity high enough to seriously test the cohesive strength of the bullet to resist the centrifugal stress; but that such is the case is indicated by the following, especially with the weaker metals like lead: For example, the twist in the United States 0.30-caliber rifle is one turn in ten inches, and the muzzle velocity of the bullet when fired with the regular charge of smokeless powder is about 2,300 feet per second. Assuming that the bullet does not strip in the rifling, it follows that it must spin at the rate of $\frac{12}{10} \times 2300 = 2760$ turns per second. The circumference of the bullet is $0.30 \times 3.1416 = 0.9425$ inch, which, multiplied by $2{,}760 = 2{,}601$ inches, or 216.8 feet per second peripheral velocity. The peripheral velocity of 216.8 feet per second, or 12,880 feet per minute, is one that would test the strength of cast-iron and rupture metal of less strength and much greater density. Hence it follows that there are two reasons at least for making high-velocity bullets of stronger metal than lead, one being to withstand the centrifugal stress and the other, and perhaps more important one, to resist upsetting in the barrel on account of the tremendous pressure required to overcome its inertia, especially where the bullet is greatly elongated, as is necessary to maintain a reasonably flat trajectory.

ON THE LOCATION OF THE PITCH CIRCLE IN WORM GEARING.

RALPH E. FLANDERS.

Mr. Perrigo* and Mr. Edgar,† in their recent contributions on this subject, have called attention to some important points in connection with this form of gearing. The writer feels, however, that the recommendations they make cannot be followed blindly, but must be applied with a full knowledge of the limitations within which these recommendations are useful. It is the purpose of the present article to point out these limitations.

Mr. Perrigo describes a worm and worm wheel which he has incorporated in the feed mechanism of a screw machine. Made in the way he describes, this worm and wheel have outlasted everything of their kind in his previous experience, and if the cases with which he mentally compares this one have no other important points of difference, his confidence is certainly justified. Unfortunately, this point is not covered, and so we are left without a solid foundation on which to base our judgment.

The feed worm of a screw machine, if it is of the class in which the worm is dropped out of engagement when the feed is released, does its work under peculiarly trying circumstances. The writer's experience in screw machine design has led him to believe that the proper proportioning of these parts is a matter of considerable importance. Consider the case of a bronze wheel and a hardened-steel worm working under the pressure of a heavy cut: When the worm is released from engagement with the wheel, under the pressure of this heavy cut, the sharp, hardened corner of the worm-tooth goes sliding down the face of its corresponding tooth in the wheel, giving it a last dig as it jumps by the corner. The necessity for quick handling demands that the momentum of the revolving

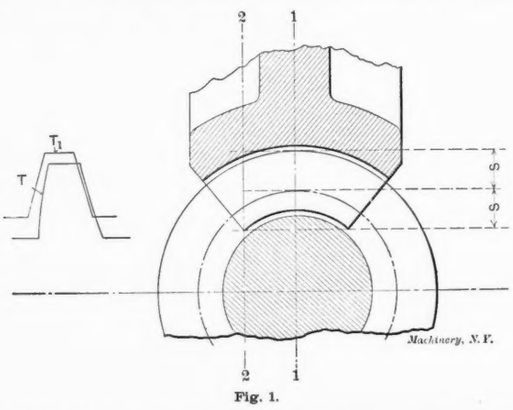


Fig. 1.

parts of the feed mechanism be kept as low as possible, so the peripheral speed of the worm wheel must be as low as possible in comparison with the rate of movement of the slide. This, in turn, requires that the worm work under heavy pressure. It is not practicable to locate the feed release between the worm and the clutch, especially if the feed is to be stopped automatically, because it is difficult to handle a toothed clutch under a severe torsional strain. Usually this problem is settled by a compromise whose success depends on the judgment of the designer; the peripheral speed of the worm wheel is made as high, and consequently, the worm thrust is made as low as is possible without too great a sacrifice in rapidity of handling. In large machines this difficulty may be overcome by connecting the pinion shaft to the worm wheel by frictional contact, accomplished by tightening up a supplementary pilot mounted in front of the main pilot wheel; the automatic release is effected by stopping the rotation of the worm.

Another point that militates against the durability of this mechanism when a releasing worm is used, is the indeterminate location of the worm. While it is obvious that a worm cannot be adjusted in a direction parallel to the axis of the worm wheel, it is not generally realized that the center distance between its axis and that of the wheel cannot be varied without losing the perfect action which exists when the worm

* An Example of Worm Gearing, MACHINERY, June, 1905.
† The Worm Gear, MACHINERY, Engineering Edition, October, 1905.

is properly located. That this is so will be evident from Fig. 1. In this cut T_1 and T are sections of a worm tooth taken on lines 1-1 and 2-2, respectively. The section on 1-1 is evidently that of an involute rack tooth and so possesses the characteristic property of correct action at any center distance, so long as its straight face is in contact with the mating gear tooth. As we leave this section, however, and approach section 2-2, the tooth outline gradually loses its resemblance to the involute form and takes a shape in which positive location is absolutely necessary for correct action, as is shown by the curved sides. This variation from the involute shape is especially marked in worms of large helix angle and consequent high efficiency.

Now, if the worm is slightly separated from its correct location in the mating wheel and no sideways motion is allowed, it will be seen by observing the relative angularity of the outlines of the faces in the curves T_1 and T_2 that the contact will at once lose its character of line contact, extending across the full width of the gear, and will be concentrated in point contact on the extreme outer edge, where correct action is impossible except at the calculated center distance. For working under heavy pressures, then, it is necessary that the worm agree in shape with the hob which cut its mate, and that its axis exactly coincide with that of the hob when it was taking its finishing cut. These requirements may be met easily in high-grade work, such as is the rule in making a worm-gear drive for a gear-cutter spindle or an elevator, but such workmanship is very far from the haphazard fitting that a releasing feed worm must necessarily get.

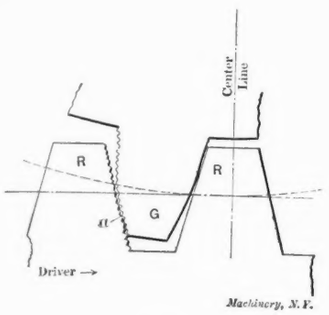


Fig. 2.

In looking over the cut of the screw machine with which the article we are discussing is illustrated, the suspicion struck the writer that the worm, or worms, must be of considerably greater helix angle than is usual in feed gearing. The unusual arrangement of a double reduction is employed, making use of two sets of worms and wheels in series. Unless the feed shaft rotates at high speed, or the feed is exceedingly fine, this must mean that the reduction in each set of gears is small, which in turn predicates a large helix angle and an efficient gear.

Mr. Perrigo must, then, give us more definite information if his experience is to be valuable as a permanent record in the matter of the location of the pitch line. Was his machine furnished with a releasing worm for a feed stop, and were the machines with which he compares it so equipped? How carefully was the worm fitted in the last machine and in the former machines? What are the helix angles of these worms and former unsuccessful cases? What materials were used in the different sets of gears which are under comparison?

Mr. Edgar has shown quite plainly that the advantage to be gained by lessening the diameter of the pitch circle on the worm is due to the fact that in such a case the contact between worm and wheel takes place for the most part after the teeth have begun to recede from each other. In Fig. 3 the worm, with its pitch line at $G H$, driving the wheel in the direction shown, will always make contact with it along the line of action, $C D$. The pitch line is located, as usual, halfway down the working depth of the tooth, and as may be seen, the contact is almost equally divided on each side of the center line. In Fig. 4, with the same reference letters, the pitch line has been located according to the rule proposed by Mr. Edgar, and the contact between the teeth is seen to take place almost wholly during the time when they are leaving each other.

Friction between two rubbing surfaces is due to the resistance imposed by the microscopic irregularities which exist on even the smoothest surfaces. In Fig. 2 are shown two teeth approaching each other, in which these irregularities are greatly exaggerated. R is the driving and G is the driven tooth. Evidently if these irregularities were as great as shown, the teeth would lock together and movement would be impossible; on the other hand, if G were the driving tooth, and the teeth were separating, there would be little to hinder their free movement. It is, then, desirable that most of the contact should take place when the teeth are leaving each other, to avoid friction, loss of power, and wear of tooth surfaces.

Fig. 1 shows the way in which Mr. Edgar proposes to locate the pitch circle of the worm. This circle is tangent to a line which lies at equal vertical distances from the extreme working points of the worm wheel tooth, and he locates the pitch line here because it is so located in a spur gear. To the writer it seems that there is no analogy between them. The pitch line of a spur gear is located at one-half the working depth of the tooth because it is required that a set of standard spur gears be interchangeable, a gear of any number of teeth meshing perfectly with a gear of any other number of the same pitch. This requirement is entirely outside of the sphere of worm gearing, so we may locate the pitch line at any point that will give favorable results as regards efficiency and durability.

The location of the pitch line affects the working qualities of the gearing in four ways, at least. With a worm of given diameter and pitch, and a wheel of given number of teeth and angle of contact, it determines the effective working area of the teeth in both members, the strength of the teeth in the wheel, the number of teeth in contact, and the nature of the contact, that is, whether it takes place during the approach or the release.

Fig. 3 shows a central section of a worm and wheel calculated in the usual manner. If a is equal to the pressure angle and angle $F D O$ is a right angle, a circle drawn from

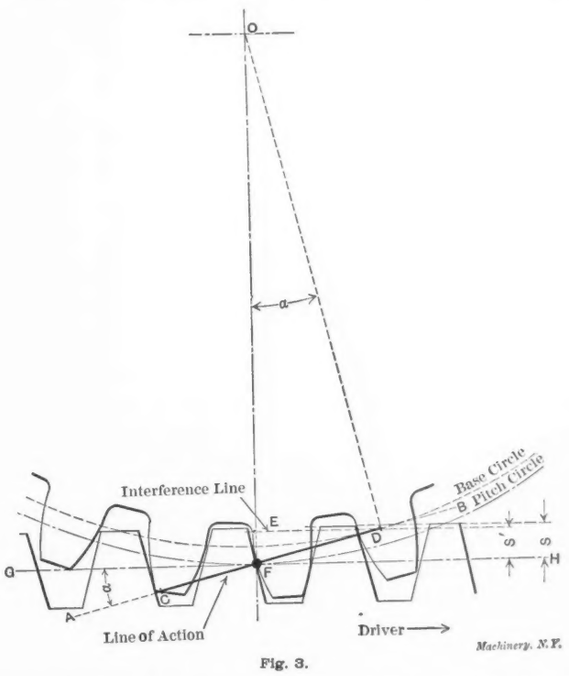


Fig. 3.

center O through D will be the base circle from which the involute curve is formed, and the line of action—the line in which the working contact between the teeth will take place—will lie in line $A B$. This line of action will evidently be limited at one end by C, the point where it crosses the outside diameter of the wheel at its throat, and at the other by D, the point where line $A B$ is tangent to the base circle, since the involute does not extend inside of the circle from which it is derived. It is plain, then, that all that part of the wheel tooth which lies inside of the base circle is clearance, and unfit for bearing surface, and that all of the rack tooth which extends above point D, or the "interference line," as it is marked, serves no useful purpose. This area of the worm tooth extend-

ing above the interference line, is seen to be slight for a thirty-tooth wheel of standard design. An inspection of the cut will show that there are always two and sometimes three teeth in contact. The contact takes place about equally each side of the center line, inclining toward the favorable side, since line $F D$, on the release, is somewhat longer than $C F$, on the approach.

In Fig. 4, we may see what effect has been produced by increasing the addendum of the worm, as we are advised to do. In the first place, the interference line, through point D, is lowered so far that the useful area of the teeth has been greatly decreased. There is no working contact on the wheel teeth inside of the base circle, nor do the worm teeth serve any useful purpose above the interference line.

The strength of the wheel tooth has been decreased. The great length of useless worm tooth extending above the interference line has cut a deeper clearance into the flank of the wheel tooth, thus weakening it at the very point where it needs strength. On pages 68, 69, and 71, of the Brown & Sharpe "Treatise on Gears," will be found three illustrations which show this point very clearly. One case is that of a twelve-tooth worm wheel of standard design, which gives a badly undercut flank. In the next illustration the worm wheel has been hobbed according to Mr. Edgar's rule, and the result is worse than in the first case. The last illustration shows the pitch line thrown clear to the outside diameter of the worm, this being advised as the proper remedy to secure a tooth of sufficient strength and bearing surface.

Referring again to Fig. 4, the number of teeth in contact has been reduced until there is only one constantly in use, though two are in position to work most of the time. The single gain to be derived in return for the advantages that have been lost lies in the fact that a greater percentage of the line of action lies on the releasing side of pitch point F than before, since $F D$ is noticeably longer than $F C$.

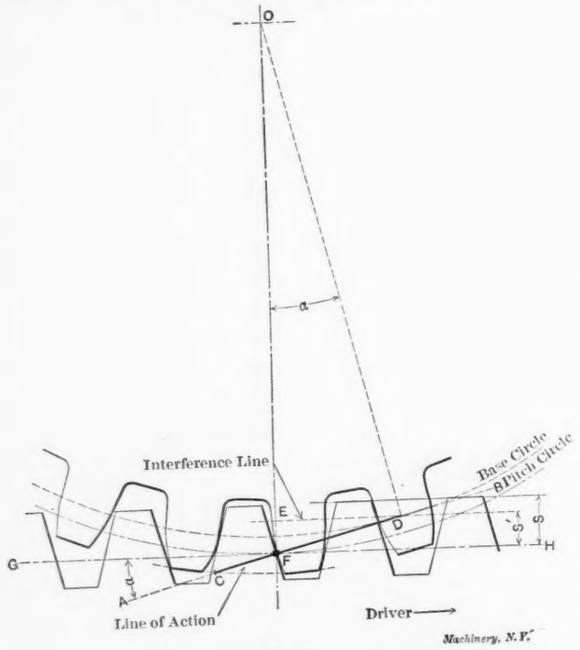


Fig. 4.

Of course only the action on the center line has been analyzed. The writer has studied the action at sections made in different places in the worm-wheel face, and it looks as though the conditions at the center line were a fairly good index of what is going on nearer the sides. The line of contact appears to rise slightly toward the outside of the worm as it leaves the center (going toward the leading side of the worm), and then drops again toward the edge of the wheel. On the retreating side of the worm the contact drops continuously. This tends to minimize the effect that the width of the wheel has on the action.

How, then, should the pitch line be located? It seems to the writer that the problem is so involved that in a case of any importance the designer should not trust to any empirical rule, but should plan each case with reference to these four points: area of bearing surface in the teeth, strength of the teeth, number of teeth in contact, and location of contact, whether in the approach or the release. To these should be added a fifth point, more important than any of the others, as far as efficiency is concerned, and that is in relation to the helix angle of the worm: it should be as large as possible.

Taking all these points into consideration, it would seem that, for worms and wheels made as they usually are for ordinary service, from ordinary materials, and with ordinary carefulness of workmanship in making and fitting, it is hardly worth while to bother about changing the location of the pitch line for the sake of having the contact on the release. It introduces too many other complications into the problem. Still, if there is any one who wants to try the effect of altering the worm and wheel dimensions with this end in view, here are a few suggestions in the shape of formulas to add to those of the two contributors who have previously written on this subject.

Let N = number of teeth in wheel.
P' = linear pitch of worm.
D = throat diameter of wheel.
d = outside diameter of worm.
D' = pitch diameter of wheel.
d' = pitch diameter of worm.
a = pressure angle.
$E = \dfrac{D' + d'}{2}$ = center distance between the worm and the wheel.
S' = Effective height of worm tooth above pitch line (see Fig. 4).

An inspection of Fig. 4 will show that S' may be expressed as follows:

$$S' = \frac{D' \ (\text{Sin } a)^2}{2}.$$

If we limit the height of our tooth to this line, thus allowing no interference, we may use the following formulas, it being considered that we have given E, P', and N.

$$D' = \frac{N P'}{\pi} \qquad (1)$$

$$d' = 2 E - D' \qquad (2)$$

$$d = d' + D' (\text{Sin } a)^2 \qquad (3)$$

$$D = D' + 1.273 P' - D' (\text{Sin } a)^2 \qquad (4)$$

For a pressure angle of 14½ degrees and an allowed interference equal to that of a standard worm in mesh with a 25-tooth wheel, these last two formulas will become:

$$d = d' + \frac{D'}{13} \qquad (5)$$

$$D = 0.923 D' + 1.273 P' \qquad (6)$$

These formulas will give as much of the contact on the release as is possible without too much undercutting; the location of the pitch line will, of course, vary widely. Formulas 3 and 4 (when $a = 14\frac{1}{2}$ degrees) are good for any number up to 64 teeth, and formulas 5 and 6 up to 52 teeth. Above these numbers the formulas would bring the pitch line below the root diameter of the worm, which is needless; so for such cases, formulas 3, 4, 5, and 6 should be replaced by the following, which will keep the pitch line within the working area of the tooth.

$$d = d' - 1.273 P' \qquad (7)$$

$$D = D' \qquad (8)$$

All that has been said in the preceding paragraphs refers only to worms whose tooth outlines show straight sides on an axial section. If, as is often the case with steep-pitched worms, the cutting tool is made with straight sides, but tipped up at an angle to agree with the helical angle of the worm, an axial section will show teeth with curved sides whose shape will depend upon the helical angle. In such a case as this it is impossible to apply any of the rules which govern the action of involute teeth, and the only way to go about the matter of locating the pitch line to suit the ideas of the designer is to make a careful analysis of the tooth action on various sections. This operation would be so troublesome and tiresome as to be impracticable under any ordinary circumstances.

TECHNICAL READING AND FORMULAS.—2.

C. F. BLAKE.

STRAINS AND STRENGTH OF MATERIALS.

Having mastered the written engineering language sufficiently to deal successfully with formulas, the next step is to make the acquaintance of such engineering terms as are most frequently met with. Foremost among these are the terms relating to the strength of materials.

If a bar is laid across two supports as in Fig. 4, and a weight placed in the center of it, we shall, if the bar be limber, witness the bending of the bar as shown, or as expressed in engineering terms, the deflection of the bar. It is obvious that the stiffer the bar, the less the deflection, and that a bar might be so lacking in stiffness as to actually break when the weight is placed upon it. Now the bar may lack stiffness from one of two causes; it may be that its dimensions are not well proportioned, or it may be made of soft and pliable materials. Sometimes both these causes are combined in the same bar

If the bar does not break when the weight is placed upon it we must admit three facts; first, that the weight bends the bar; second, that the bar resists the bending; third, that the bar is able to resist the bending because it is large enough and made of stiff enough material.

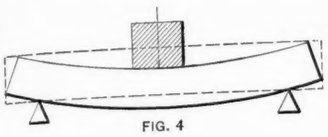


FIG. 4

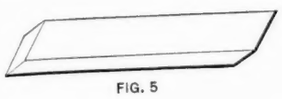


FIG. 5

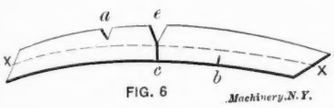


FIG. 6 Machinery,N.Y.

The bending effect that the weight has upon the bar is called the bending moment upon the bar due to the weight.

The ability of the bar to resist the bending is called the moment of resistance of the bar.

How these names first came into use the writer does not know, perhaps there is no explanation, but the reader must not confuse the terms with any period of time because of the word moment. Time has nothing whatever to do with the strength of the bar, or the effect of the load upon it.

In Fig. 4 the point at which the greatest bending occurs is directly under the weight, and we say the bending moment is maximum at this point, and the moment of resistance of the bar must equal the maximum bending moment at this point in the bar.

In using the term bending moment, the engineer usually means the maximum bending moment, because this is the greatest bending effect upon the bar, and we shall hereafter drop the word maximum.

If now we let M = the bending moment on the bar, and R = the moment of resistance of the bar, we can express the relation of the two as given above thus:

$$M = R. \qquad (4)$$

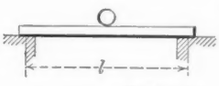
We said that the maximum bending moment was under the weight, and if the weight is placed further along on the bar, nearer one support than the other, the maximum bending moment will move with the weight. Also if the bar is differently supported the maximum bending moment will be at another point.

For all cases of frequent occurrence engineers have tables of formulas giving the position and amount of the maximum bending moment, so that it is only necessary to find in the tables the same case as the one we are considering, and taking the formula there given, substitute for the letters the corresponding dimensions in our case, and we have a numerical expression for the bending moment. The formulas given in these tables consist of combinations of dimensions measured along the bar, and weights of the loads on the bar. If when substituting values for letters in the formula, loads are taken in tons, and distances in feet, the bending moment will be expressed in foot-tons, while if loads are taken in pounds and distances in inches, the usual custom, the bending moment will be expressed in inch-pounds.

The following is a small portion of such a table as may be found in any book on machine design in any drafting room or factory, as well as in all the hand books issued by the steel mills.

TABLE 3. BENDING MOMENT OF BEAMS UNDER VARIOUS SYSTEMS OF LOADING.

W = total load.
l = length of beam in inches.
I = moment of inertia.
Z = section factor.

Case I.		Beam fixed at one end and loaded at the other. Max. bending moment at point of support = $W l$.
Case II.		Beam supported at both ends. Single load in middle. Max. bending moment at middle of beam $\frac{W l}{4}$.

Machinery,N.Y.

So much for the first member of our equation, the bending moment on the bar. We have already seen that the bar offers resistance to bending by reason of two things, its dimensions and the character of its material, and we should expect to find both dimensions and material accounted for in the formula for the moment of resistance of any bar. This is just what the formula for the moment of resistance does. It is composed of two parts or terms, one of which expresses the resisting effect of the material of the bar, and the other expressing the resisting effect possessed by the bar because of its shape and size. Let us investigate each term by itself, taking first the resisting effect of the material.

Let the reader take an ordinary rubber eraser of the form shown in Fig. 5, and bend it as shown in Fig. 6. While holding the eraser in the bent position, draw a sharp knife across the top side. The cut immediately spreads out in the form of a V as shown at *a*. Draw the knife a second time through the same cut and the V spreads a little more. Now draw the knife across the bottom. The cut immediately closes up as at *b*. Draw the knife a second time across the same cut and it will still close up completely. In making the second cut on this side it may be necessary to release the eraser from the bent position because the closing cut grips the knife blade and makes cutting difficult, but the cut will close, upon again bending the rubber.

Having made the two cuts *a* and *b*, reverse the bend in the eraser and witness the closing of cut *a* and the opening of cut *b*. Now if you are a careful experimenter, you can start two such cuts as *a* and *b* directly opposite each other, and by cutting each one the same amount each time, you can succeed in bringing them nearly together in the center as shown at *c*. Of course it will be impossible to bring them quite together because that would cut the eraser apart, but by a little care you can satisfy yourself of these facts; that the portion of the eraser above the center line *x x* separates when cut; and that the portion below the line closes when cut. Reversing the bend of the eraser as before reverses the behavior of the cuts, but observe that whichever way the eraser is bent, the opening cuts are to be found on the convex side, and the closing cuts on the concave side.

We know that all material (engineering and building material at least) is composed of fibers, and we must conclude from the behavior of our eraser that all the fibers on the convex side of the line *x x* are stretched when the eraser is bent, while the fibers on the concave side of *x x* are compressed.

Since the cut through the stretched fibers opens like a V we may conclude that those fibers lying at the top of the V are stretched more before the cutting than those lying at the point of the V.

A careful examination of the cut made through the compressed fibers will show that at the outer portion of the cut, the edges are raised slightly, while at the inner portion near the center of the eraser, the edges are not raised. We can account for this only by assuming that the fibers at the outer portion are more compressed than those near the center of the eraser.

Having performed these experiments and noted the results, we must admit the following facts: 1st, that half the fibers of

a bent bar are in compression while the other half are stretched, or as engineers say, are in tension; 2nd, that the amount of compression or tension is greatest at the outer portion of the bar, and diminishes towards the center of the bar; 3rd, that it follows from this, as well as from experiments with cut *c*, that there must be a line through the center of the bar where the fibers are neither in compression nor tension.

Now the fibers resist any change in their condition, either stretching or compressing, the amount of resistance differing in different materials. Iron fibers resist more than rubber fibers for instance.

When a bar is bent, engineers speak of the fibers as being under stress, some being under compressive stress, and others under tensile stress, as we have seen, and they speak of the bar as being subjected to fiber stress.

Now fiber stress is expressed in pounds per square inch, and it is the duty of the engineer when designing a beam or other structure to keep the fiber stress within safe limits, and these safe limits are given in hand books for a great variety of materials, in tables of which the following is a sample.

TABLE 4. STRENGTH OF MATERIALS—POUNDS PER SQUARE INCH.

MATERIALS.	Ultimate Strength.	Safe Working Strength.	Factor of Safety.	Kind of Stress.
Cast Iron.	88,640	16,000	5	Compression
	15,620	3,000	5	Tension
Steel.	82,500	16,000	5	Compression
	80,000	16,000	5	Tension

Since the material composing the bar derives its ability to resist bending by reason of the resistance of its fibers to changes, the fiber stress must be one of the terms expressing the moment of resistance of the bar. The fiber stress is denoted by the symbol *f*.

The second term of the moment of resistance of the bar takes into consideration the strength the bar derives from its dimensions.

Bend the eraser in the direction of its greater thickness. We note it takes a much greater force to bend it thus than to bend it as we did at first in the direction of its least thickness.

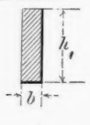
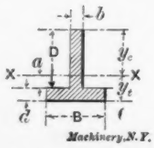
If we repeat the experiments with the cuts while bending the eraser thus, we shall find that everything witnessed before holds good for this case also.

If we look for a reason for the greater force required to bend the eraser in the direction of its greater thickness, we shall find it in the fact previously observed, that the fibers are more stretched or compressed the further they are from the center line $x\,x$, and thus they present greater resistance to bending.

The line $x\,x$ is called the neutral axis, because on it the fibers are neutral, being neither stretched nor compressed, and the fibers at the outer portion of the bar are called the extreme fibers, because they are furthest removed from the neutral axis $x\,x$.

The second term of the moment of resistance, taking account of the shape and size of the bar, is called the section factor, sometimes also called the section modulus, and is given in all hand books in the shape of tables for different shapes of beams, like the following sample.

TABLE 5. VALUES OF I (MOMENT OF INERTIA) AND Z (SECTION FACTOR) FOR VARIOUS SECTIONS.

	Section.	I	Z	Area.
Case I.		$\frac{b\,h^3}{12}$	$\frac{b\,h^2}{6}$	$b\,h$
Case II.		$\frac{b\,y_c^3+By_t^3-(B-b)a^3}{3}$	$\frac{I}{y_t}$ $\frac{I}{y_c}$	$Bd+bD$

Machinery,N.Y.

The neutral $x\,x$ is not always in the center of the bar, but it always passes through the center of gravity of the cross section of the bar.

Here we shall have to digress for a moment, since in writing these papers it is the intention to leave no term unexplained. The reader may best become acquainted with the center of gravity in the following manner: Cut out of stiff cardboard the shape of the cross section of the bar, and balance it over a sharp edge, such as a triangular rule. Draw a line across the card corresponding to the edge over which it is balanced. Repeat the experiment, turning the card around on the edge, and balancing it a second time, draw another line. The intersection of these two lines will be the center of gravity of the section of the beam. If the experiment has been done with sufficient care, the card may be balanced upon a sharp point placed at the intersection of the two lines, just as if the entire material of the card were placed vertically above the point. A definition frequently met is: The center of gravity is that point at which the entire weight of a body may be considered as concentrated.

Another way of finding the center of gravity is to suspend the card by a fine thread alongside of a plumb line, and when the card and line have come to rest, mark the position of the plumb line on the card. Turn the card around and suspend a second time from a different point, and mark the position of the plumb line again. Where the two marks of the plumb line cross will be the center of gravity of the figure. No matter from how many points the card may be suspended, the plumb line will always be found to pass through the center of gravity. A line on the side of the beam directly opposite the center of gravity thus found will be the neutral axis. See Fig. 7.

If we now take the equation expressing the relation of the bending moment on the bar to the moment of resistance of the bar, and use the symbols for the two parts of the moment of resistance, we shall have

$$M = R = f\,Z.$$

Some tables do not give the section factors Z for all sections directly, but say it is $\frac{I}{y}$, and therefore we must understand this expression.

The denominator y of the fraction is the distance from the neutral axis $x\,x$ to the extreme fiber of the bar, see Fig. 8, and the numerator I is what is called by engineers the moment of inertia of the section of the bar. Here again there is a chance for confusion because of the use of the word inertia.

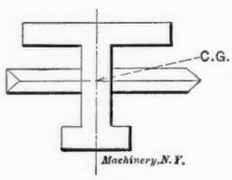


Fig. 7.

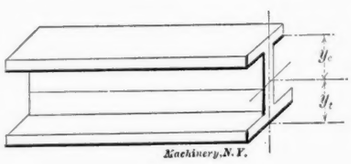


Fig. 8.

"The term moment of inertia was originally employed when comparing the energies of rotating bodies. We know that a moving body possesses energy due to that property of matter which engineers call inertia. Inertia is not a force; it is simply resistance, and is due to the incapability of a dead body to move, or of a moving body to change its velocity or direction without the application of some external force. Now the number of foot-pounds of energy possessed by a moving body is equal to $\frac{1}{2}\,M\,V^2$, where M is the mass of the body, and V its velocity in feet per second. A moving body then, must be acted upon by an external force before it can be brought to rest. A rotating body is simply a very large number of particles moving in circular paths about an axis called the axis of rotation. Each moving particle, therefore, possesses energy due to its inertia, and the energy of each particle is equal to $\frac{1}{2}\,m\,v$, where m is the mass of the particle, and v its velocity in feet per second. But the energy varies as $m\,v^2$, because simply dividing by 2 does not change the relative values. It is also obvious that the circumferential velocity of each particle varies as the distance from the axis of rotation, which distance or radius we call r. Hence, substituting r for v, the energy of each particle varies as $m\,r^2$. Suppose we imagine that the whole mass of the rotating piece, that is the sum of all the small particles m, is concentrated in a circle that is of

such diameter that the energy possessed by the entire mass is the same as before. The radius of this imaginary circle is called the radius of gyration, and is usually designated by the letter r. Now we may say that $M r^2$, where M stands for the whole mass, is a measure of the energy of the rotating piece. This expression $M r^2$ is given the name moment of inertia, each particle of which the rotating body is composed possessing a turning moment about the axis of rotation, due to its motion and inertia.

When it was discovered that the flexure of a beam depended upon the value $a r^2$ (where a is the area of the cross section of the bar, and r^2 is the mean of the squares of the distances of the infinite number of small areas into which the area of the section may be supposed to be divided, from the center of gravity of the section) $a r^2$ was seen to be similar to the expression $M r^2$, which in connection with the rotating bodies, had already become known as the moment of inertia; so, very carelessly on the part of those who first committed the error, it was said that the flexure of a beam varied as its moment of inertia, not because inertia has anything to do with it, for of course it has not, but because $a r^2$, the expression for the moment of resistance to flexure, happened to vary in the same way as the moment of inertia $M r^2$ of the same body when rotating about its center of gravity."

The moment of inertia of a bar may be calculated by several methods, but the tables give it for all usual shaped sections, and we will not attempt the calculation in these papers.

Since we are sometimes able to find in tables only the moment of inertia of a bar, and not the section factor, we must bring our formula one step further, thus:

$$M = R = f Z = f \frac{I}{y} \qquad (5)$$

or

$$Z = \frac{I}{y} = \frac{M}{f} \qquad (6)$$

and here we have the formula for determining the size required for any beam.

For beams in which the center of gravity is not in the center of the beam, there will be two values of y, one of which we will denote as y_c being the distance from the neutral axis to the extreme fibers in compression, and the others as y_t being the distance from the neutral axis to the extreme fibers in tension, see Fig. 8.

In some materials the ability of the fibers to resist tension is about equal to their ability to resist compression, while in other materials there may be great inequality in this direction, some being much stronger in tension than in compression, while others are stronger in compression than in tension. In such a material we shall have two values of f, which we will denote as f_c and f_t for compression and tension respectively.

Some tables on the strength of materials give what is called the ultimate or breaking strength of the materials, while other tables give the safe working strength of materials.

When using the latter tables, the values given are to be substituted directly for f_c and f_t in the formula. Since it would not do to have the material of which a beam is made strained up to the breaking point, we must, when using the former tables, make use of a factor of safety. This factor of safety is a divisor by which the breaking strength of a beam is divided to allow a margin of strength in the beam. This divisor varies from 2 to 10, and the proper use of different divisors is given in the text books.

To illustrate, the breaking strength of steel may be given as 80,000 pounds per square inch, and $\frac{80{,}000}{5} = 16{,}000$. If we substitute 16,000 for f in the formula, we shall be working our material with a factor of safety of 5, and the beam should not actually break until loaded with five times the load designed for. As a matter of fact, the beam would become badly bent long before five times the load could be placed upon it.

We have seen that all material deflects under the influence of a load, and we suppose that the elasticity of the material causes it to spring back to its original condition when the load is removed. This is true within limits, but there is a point somewhere between the safe load and the breaking load at which, when the load is gradually increased, the beam becomes strained beyond its power to return to its original condition upon the load being removed. This point is variously called the limit of elasticity, the yield point, the point of permanent set.

Let us now take up two examples illustrating the ground we have just passed over, and the use of the tables.

Example 1. A rectangular steel bar 2 inches thick is built into a wall as in Fig. 9, and is to hold a load of 3,000 pounds at its outer end, 36 inches from the wall. We wish to know the required depth to make the beam.

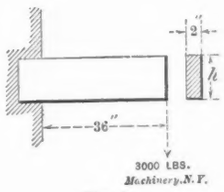


Fig. 9.

1st, consider the bending moment on the beam. According to case 1, table 2, the bending moment is

$$M = W l.$$

For our case we know W and we know l, and substituting these for the letters in the formula gives us

$$M = 36 \times 3{,}000 = 108{,}000 \text{ inch-pounds.}$$

2d, consider the permissible fiber stress in the steel bar. Table 3 gives the safe working strength of steel as 16,000 pounds per square inch.

3d, using formula (6) we can find the value of the section factor for our beam. We know the bending moment and we know the fiber stress, so substituting these for the letters in the formula we get

$$Z = \frac{M}{f} = \frac{108{,}000}{16{,}000} = 6.75.$$

4th, find the section of our beam in table 4, case 1, where we find that the section factor is

$$Z = \frac{b h^2}{6}.$$

We know Z and we know b, so substituting these values for the letters, we get

$$6.75 = \frac{2 \times h^2}{6}.$$

If we multiply both sides of this equation by 6, we shall not change its value, but shall get

$$6 \times 6.75 = 2 \times h^2.$$

If we now divide both sides by 2, we shall not change its value, but shall get

$$\frac{6 \times 6.75}{2} = h^2 = 20.25$$

5th, we can most conveniently find the square root of 20.25 from a table of squares and roots which may be found in any hand book. This square root is 4.5, and we thus find that

$$h = 4.5 \text{ inches.}$$

If we make the beam 2 inches thick by 4.5 inches deep by 36 inches long, it will support a load of 3,000 pounds at its free end, and the fibers will be strained to 16,000 pounds per square inch.

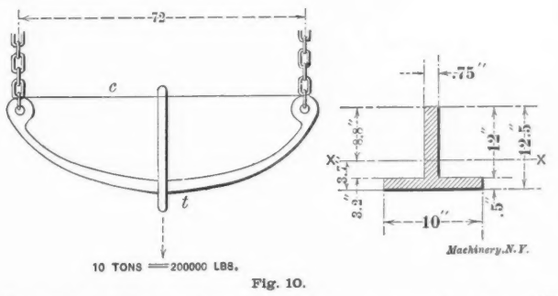


Fig. 10.

Example 2. Let us have to design a suspension beam like Fig. 10 to carry ten tons, the material to be cast-iron. The proposed section of the beam is more complicated than that of the previous example, and we cannot obtain a result quite so directly.

1st, inspect the proposed beam to locate the compression and tension flanges. We find the compression flange is on top and

the tension flange on the bottom, and we mark them c and t respectively.

2d, table 3 shows us that cast-iron is stronger in compression than in tension, hence we conclude we should have more metal on the tension side than on the compression side, and accordingly we place the section with the heavy side down.

3d, assume a section by making the best guess possible as to the dimensions shown heavy in the figure. Cut this section out of cardboard, and find the location of the neutral axis $x\,x$ as previously explained. Now fill in the figures shown light by measuring the cardboard section.

4th, find the section in table 4. Here we find that before we can get the section factor of the beam we must get the moment of inertia of the beam. Substitute the dimensions of our section for the letters of the formula given in table 4, and we shall get

$$I = \frac{(.75 \times 8.8^3) + (10 \times 3.7^3) - (10 - .75)\,3.2^3}{3}$$

$$= \frac{511.1 + 506.5 - (9.25 \times 3.2^3}{3}$$

$$= \frac{1017.6 - 303.12}{3} = \frac{714.48}{3} = 238$$

5th, now divide the moment of inertia just found by the distances of the extreme fibers from the neutral axis, that is, by y_c and y_t, and we get

$$Z_c = \frac{I}{y_c} = \frac{238}{8.8} = 27 \text{ section factor for compression side.}$$

$$Z_t = \frac{I}{y_t} = \frac{238}{3.7} = 64.3 \text{ section factor for tension side.}$$

6th, inspect table 2 and find the bending moment on the beam according to case 2, substituting the dimensions of the beam and the load to be carried in the formula given, we have

$$M = \frac{W\,l}{4} = \frac{20{,}000 \times 72}{4} = 360{,}000 \text{ inch-pounds.}$$

7th, dividing the bending moment just found by the section factors found in the 5th step, will give the fiber stress on the beam according to formula (6), thus

$$\frac{360{,}000}{27} = 13{,}333 \text{ pounds per square inch on the compression side,}$$

$$\frac{360{,}000}{64.3} = 5{,}600 \text{ pounds per square inch on the tension side.}$$

The latter is too high, so another guess must be made, making the section heavier on the tension side. Then the steps 3, 4, 5 and 7 must be repeated, and if the fiber stress then comes below 3,000 pounds per square inch, the section will be right.

* * *

HOW A KNIFE-EDGE STRAIGHTEDGE WAS MADE.

F. N. GARDNER.

Except among gage and fine toolmakers the use and value of the knife-edge straightedge is not so well understood by the machinist trades as it should be. Many a workman who has just turned in a nice "fine-haired" job would be surprised to have an old-timer go over it with his "knife-edge" and show him hollows that "you could drive a hack under." The knife-edge certainly will show up imperfections, and likewise magnifies them.

Some twenty-five years ago, starting out on a manufacturing venture (Hartford Tool Co.), we took up among other things the building of knife-edge straightedges. So far as my information goes, this was the first attempt to put them on the market as a regular line. It was decided to put them out, three in a set 3, 4½ and 6 inches long, together, of course, with a test-bar, otherwise how could you tell whether they were any good or not. So far everything was easy, and it was up to me to "deliver the goods." I put in several solid weeks teaching myself how to make straightedges and test-bars, and at the end was able to turn out satisfactory sets at a reasonable cost.

The blanks were milled from annealed tool steel to about the section shown in Fig. 1. The hardening was done by dipping the edge ¼ inch deep in red-hot lead and plunging in water, back first. I do not recollect any difficulty from springing. They were tempered on hot sand to a light straw color. The only machine grinding, except polishing, was done on a very simple fixture in a universal grinder, as shown in Fig. 1. This gave a sharp edge of about 60 degrees angle which was approximately straight. They were then ready for hand lapping.

The test-bars were the next proposition. We tried many and various things with unvarying poor results; hardened tool steel, casehardened soft steel, chilled castings, hard bronze, etc. Some were too hard and expensive to work and all were unreliable when the weather changed. We finally settled on plate glass as the best available material. Plate glass about

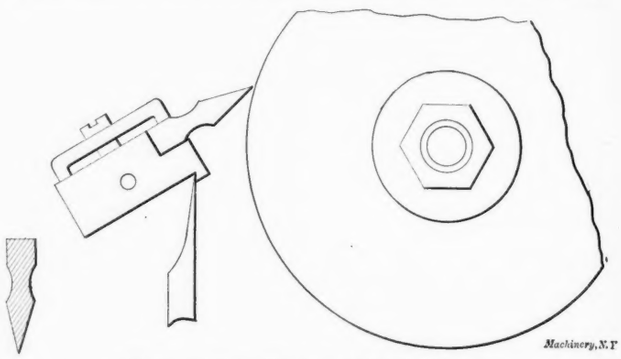


Fig. 1. Section of Blade and Method of Grinding.

5-16 inch thick was scratched with a diamond and broken into pieces 7 inches long by 1⅝ inches wide. These were ground at edges and ends reasonably square, straight and uniform in size on a grindstone by hand one at a time. The glass bars were then cemented together, with patternmakers' shellac varnish, into blocks of convenient size, say 4 or 5 by 7 inches. See Fig. 2.

The blocks, each one made up of 8 or 10 bars, were ground or lapped by hand on a cast iron plate 30 inches square with loose corundum. One side only of the blocks was thus ground down, as one true edge only was required on the finished bar. For this operation we started in with coarse grain, about No. 42, using finer grain as the grinding progressed, and completing the block lapping stage with about No. 80 or 90. The blocks were then soaked in water until they came apart, which they did very readily as only a small quantity of shellac was

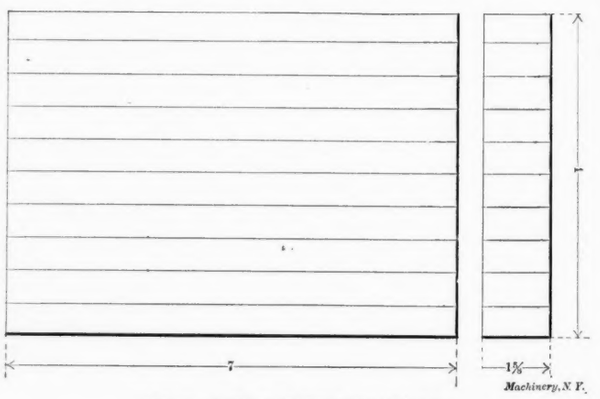


Fig. 2. A Block of Glass Straightedges.

used and that only in spots at the ends. The corners of the lapped edge were then beveled 1-32 inch all around on the lapping plate. They were then given a heavy coat of black asphaltum varnish all over except the finished edge. This seemed to completely absorb the light and made the finished edge look like vulcanized black rubber. The asphalt was covered with several thicknesses of manila paper, and then neatly bound with Russia leather, leaving the finished, or rather the to-be-finished, edge uncovered. The test bars were now ready for lapping. So far everything was comparatively easy, rapid and not very expensive. But right here I had to take up a course of manual training with myself for instructor. Re-

member it was not up to me to make one good straightedge only, but to turn them out by the hundreds at a cost price that would permit selling them at a profit.

No one would wish to follow me through all my mistakes and failures even if I could remember them, but I will try to describe some of the methods that led to such success as I attained and to indicate a few ways of "how not to do it," which are quite as necessary to understand and much more expensive to follow. I soon decided that the personal requirements were a clear eye, a steady hand and an amount of patience and perseverance that would make a Jap turn green with envy. For a lapping plate I used at first a block of fine soft cast iron 10 inches square by 3 inches thick, planed as nearly perfect as possible on the theory that if the lap was straight the work finished on it would be straight, but it didn't work out that way. It is difficult if not impossible to file a surface of any considerable size accurately straight and flat with a perfectly straight and flat file. I proved to my own dissatisfaction that it was equally difficult to lap a piece straight on a flat lap. My next move was to get a piece of cast iron 2 inches square by 12 inches long. This was accurately planed to the cross section shape shown at Fig. 3. This gave me four working sides, each of which I charged with different grades of corundum.

The method of charging this cast iron bar and the results obtained with it I consider to be the most essential features of my success. The only abrasive I used for the finish lapping was flour corundum separated by washing into four different grades. The washing method of getting abrasives down fine is so well-known that I don't need to describe it here. To charge the bar, Fig. 3, I sprinkled the flour corundum, of whichever grade I wished to use, quite liberally on the 10-inch square iron plate, laid the bar on it and rubbed it in. After a few treatments the section of the bar changed to the shape shown in Fig. 4 only not so much, of course, as shown in the exaggerated cut. The tendency was for the bar to be rounding (convex) and the plate to be hollowing (concave). Remember I am talking about small measurements—probably the difference between the highest and lowest point of the surface of the bar anywhere would not exceed five ten-thousandths of an inch, but half a thousandth of an inch opening lets a big chunk of light through under a knife-edge. When the bar was lapped until all its working surfaces were slightly rounding (convex), it was also thoroughly charged with corundum grains bedded into the metal. All loose grains had to be carefully dusted off—nothing is more destructive to your last touch of accuracy and finish than loose grains of abrasive skating around on the lap.

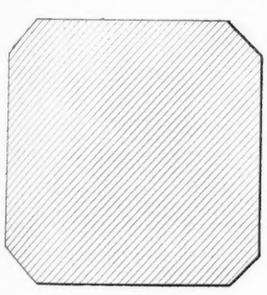
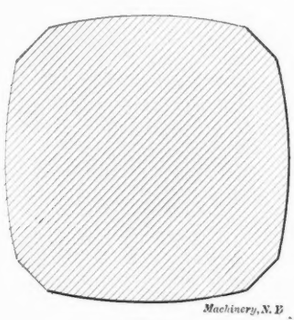


Fig. 3.



Machinery, N. Y.

Fig. 4.

I now had my tools working all right, and by that time I had learned a good deal about using them. The straightedges as they came from the grinder had a sharp wire-edge. This edge was then rounded to about 5-1,000 to 8-1,000 inch radius (free hand work and eye measurements) with an oil-stone. They were then ready to be made straight.

By days of solid labor I had acquired a "test bar" as I called it. I was as familiar with its imperfections and vagaries as a barber is with his razor. I knew its straight places and its big hollows and its little hollows, how it would act when the thermometer stood at 95 and how when it stood at 60 deg. It was made of chilled white iron. I always used it when finishing straightedges in preference to the nearly perfect glass test-bars that I was afterwards able to make. I found that its very imperfections, *i. e.*, the big and little hollows, were a help in detecting and locating the imperfections on the straightedges. The method of testing was to slide the straightedge lengthwise on the test-bar. If the hollows in the bar showed just the same under all parts of the edge, it was good evidence that the edge was straight. Of course both were held between the eye and a strong light. Light under these conditions, when it can be seen at all, seems to be greatly magnified and I always thought that I could detect a smaller imperfection in the straightedge with my old humpy iron test-bar than I could with the much more perfect glass ones.

After days many and full of trouble, I discovered something, a very simple thing, one that a person possessed of common sense ought to have known without being told. As hollows cannot be raised by lapping, I went at the high spots, but strange to say the more I rubbed the higher they seemed to grow. I thought to myself—"lap don't cut—needs charging." I take five minutes charging the lap and fifteen more starting a man on a job, come back, pick up the same piece and find a big hollow where the high spot was. The explanation was simple. The friction of lapping generated enough heat to expand the piece faster than I ground it off, and when it came back to an even temperature it showed that I had taken off too much. I suppose that the reason why I did not find out this point sooner and why others have been fooled in the same way is that I used great care *not* to heat the pieces by friction, using a slow motion and light pressure, and could not feel any heat with my fingers. At first I thought the pieces were changing and springing on their own hook—or call it "molecular adjustment"—and would finally get settled down and keep quiet. I finally got down to this procedure:

I would pick up a piece, if a straightedge would try it on the test-bar, locate the highest high spots, give them a few *swift* rubs on the rounding lap right in the place where it would do the most good, and without a second look lay it down at the back of the pile to cool off. Say there were 100 pieces in the pile, or row; in this way I went straight through the row from the 1st to the 100th and then over and over again. The high spots get lower and the hollows begin to disappear, and I to lap slower and lighter. Near the finish I used a wooden holder on the straightedges to keep the heat of my fingers away from them; some sort of a non-conducting holder should always be used on the longer straightedges—the glass test bars are sufficiently protected by their binding.

The final test, and I think it was a very severe one, was to place a fiber of cotton across one end of the test-bar, rest the point of a 6-inch straightedge on the fiber and let the back end touch the bar. If both bar and edge were accurate they would show a gradually diminishing streak of light from the fiber to within less than 1 inch of the back end of the straightedge. I was not able to measure this fiber of cotton very accurately with an ordinary micrometer, but I concluded that they averaged about 1-10,000 inch diameter. If this is approximately correct, I think we were not far off, in our catalogue statement that light could be seen through an opening 1-50,000 inch between a knife-edge straightedge and a test-bar. This statement was afterwards disputed by a correspondent in a mechanical paper and I believe has been withdrawn from the catalogue of our successors in the business. It would be interesting, to me at least, to get the opinion of men who know, on this point.

To anyone who wishes to make knife-edge straightedges or knife-edge square blades and test-bars, I will offer the following suggestions:

Use carborundum on the lap—it is better than corundum. I do not know of anything better for the lap than fine-grained soft cast iron. They do not need to be so large as the sizes I have given, much smaller pieces will answer for home-made work. Keep the surface of your bar lap slightly rounding (convex). Do not rub your work on the charging plate, but only on bar lap, and do not rub lengthwise, *i. e.*, from end to end, but hold it at an angle with the bar, say from nearly straight to square across, and be very careful to touch nothing but the high spots. There is no law against using a slip Arkansas stone on the bad high spots in the straightedge, but

if you do you must be careful. The straightedge ought to be correct when leaning anywhere to 15 degrees either way as well as when straight up and down. You will find it comparatively easy to make it shut light if held in exactly the right position, but to give it an eighth of a turn either way and still have it shut light is not so easy. Do not start out with the idea that you can work steadily on a straightedge or test-bar until it is finished. My experience is it cannot be done, but that you can do only a very little work on each piece at a time, and the nearer you get to perfection the less you can do at each handling. Make up your mind to take anywhere from a month to a year to finish a set. If you have patience and sharp eyes and learn by experience as you go along, you will have put in twenty or thirty hours' time of actual labor and have a set of tools that you will be proud of, and one that it would take a lot of love or a good bunch of money to buy.

Closing a reminiscence will say that I exhibited one of our first commercial sets to the Brown & Sharpe Mfg. Co. in their office at Providence. After examining them very carefully with Mr. Viall, Mr. Sharp turned to me and said, "We have only one fault to find with these tools—your price is altogether too low." It did not take us very long to find out that there was "much of wisdom" in this remark.

* * *

CONDENSING PLANTS FOR HIGH VACUUMS—2.

In the March, 1906, number of MACHINERY was a description of one of the 500-kilowatt Curtis turbine units of the Newport, R. I., station, which was one of the first of these machines installed. This turbine was equipped with a Wheeler condenser and an Edwards air pump made by the Wheeler Condenser and Engineering Co., New York, and the construction of this pump is such as to make possible a simple arrangement of the condensing apparatus.

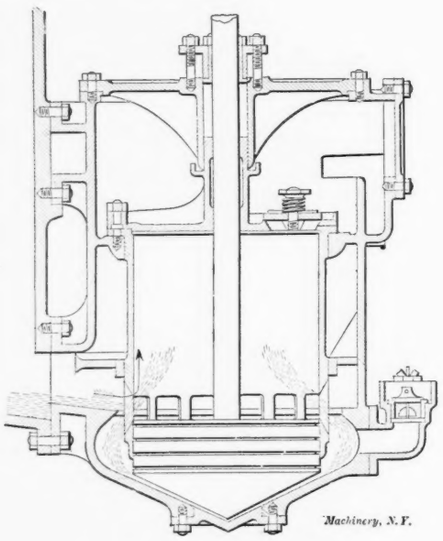


Fig. 6. The Edwards Air Pump.

Fig. 6 is a section of the pump cylinder. It is constructed without foot valves, which require a pressure in the condenser sufficiently above that in the pump to lift the valves, and overcome the inertia of the water. The condensed steam flows continuously by gravity from the condenser into the base of the pump, and is there dealt with mechanically by the conical bucket working in connection with a base of similar shape. Upon the descent of the bucket, the water is projected at a high velocity, through the ports, into the working barrel. The plunger then rises, closing the ports, and sweeps the air and water before it, causing them to escape through the valve at the top of the barrel. The elimination of the foot valves in this pump enables it to maintain a slightly higher vacuum than with the old-style pump, so that 27 or 28 inches can be maintained without the use of an auxiliary air pump.

Fig. 7 is a plan view of this turbine, condenser and pumps, and shows this to be about the most compact arrangement that can be had if a separate surface condenser is employed. By way of contrast we have, in Fig. 8, the lay-out of the apparatus shown in Fig. 3 of the last number. The condenser is located in the base of the turbine and does not appear in Fig. 8, but the auxiliaries are so disposed that the total floor space occupied is considerable.

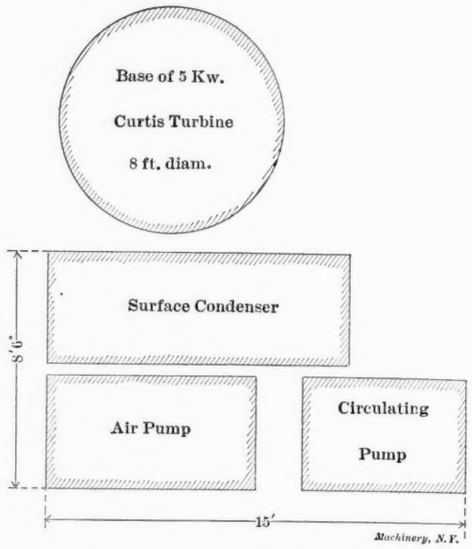


Fig. 7. Floor Plan of Newport Installation.

A novel arrangement for maintaining a high vacuum is the "vacuum augmenter" employed by Parsons with considerable success in England. He uses air pumps, as shown in Fig. 9, placed below the level of the condenser and in addition to the usual pipe connection between the air pumps and the condenser, there is a small pipe leading to an auxiliary condenser, generally having about one-twentieth the cooling surface of the main condenser. In a contracted portion of this auxiliary pipe is a steam nozzle which discharges a jet of steam that acts similarly to the jet of an injector. This jet draws nearly all residual air and vapor from the condenser, and delivers it to the air pumps. The main pipe leading to the air pump is so curved as to form an air seal which prevents the air and vapor returning to the condenser. With this arrangement there need be a vacuum in the air pumps of only about 26 inches, while the vacuum augmentor will increase the vacuum in the condenser to 27 or 28 inches. Mr. Parsons states that the quantity of steam required for this

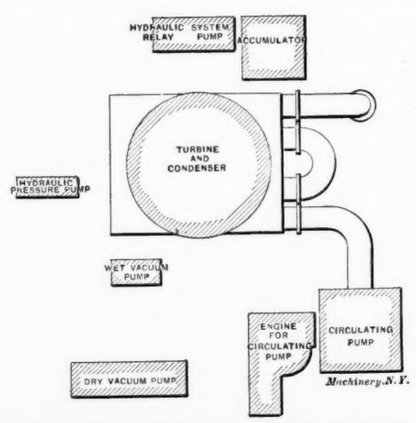


Fig. 8. Floor Plan of Unit shown in Fig. 3, October Issue.

steam jet is about 1½ per cent of that used by the turbine at full load, and this, together with the air extracted, is cooled by the auxiliary condenser.

Jet and Injector Condensers.

The surface condenser has come into extensive use with the steam turbine because the steam discharging from a turbine is entirely free from oil and if collected and condensed can be used over and over in the boilers. The feed water leaves the hot well of a surface condenser operating at high vacuum

at nearly 100 degrees F., and passes through a heater where the temperature is raised still further by steam from the auxiliaries. In jet and injector condensers the condensed steam passes off with the injection water, which is at a temperature of 80 or 90 degrees, and when part of this is used for boiler feed there is a loss of some 10 or 20 heat units per pound, as compared with the surface condenser. This is so slight, however, that it does not pay to install a surface condenser and attending apparatus on the score of heat saved. The jet type is cheaper, simpler and works as well or better.

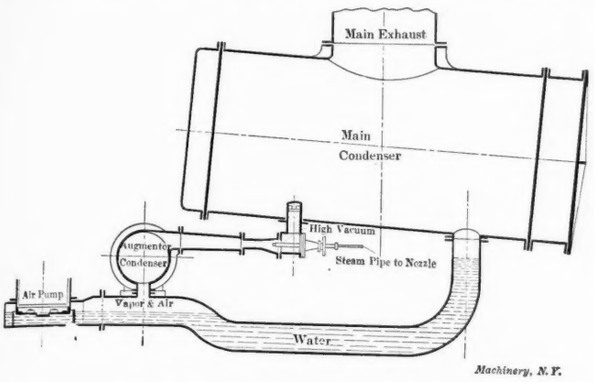


Fig. 9. Parsons' "Vacuum Augmenter."

In localities where the available water supply contains sulphate of lime, acid, grease, or other harmful impurities, or where the cost of pure water is high, the surface condenser should probably be given the preference, though if it is merely the cost of the water that is at stake, the problem should be gone into very carefully before deciding.

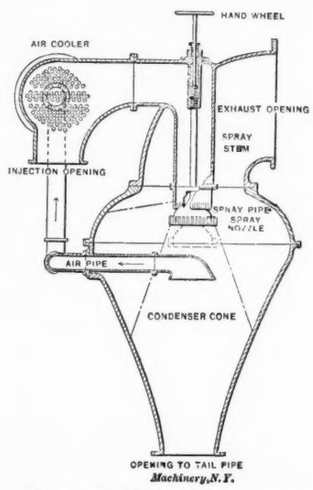


Fig. 10. Worthington Jet Condenser.

The only "out" about the operation of the jet or injector condenser at high vacuum is that more or less air is entrained with the injection water and unless this be removed it is difficult to maintain the vacuum. This can be done, however, by the use of a dry vacuum pump and an air cooler.

In a paper before the American Society of Mechanical Engineers in December, 1904, George I. Rockwood describes the installation of a Bulkley injector condenser in connection with a Westinghouse-Parsons turbine. The injection water is elevated into a vertical tank 30 inches square by 15 feet deep, in which the water level is maintained 6 inches below the water inlet nozzle of the condenser. The injection pipe takes the water from near the bottom of the tank. The air entrained with the water rises to the top of the tank and is largely eliminated from the injection water entering the condenser The flow of the injection water through the throat of the condenser is what constitutes the air pump, and it is found to be the only air pump needed, since a vacuum of 28½ inches has been maintained, regardless of whether steam is passing through the turbine or not.

A jet condenser, which is a modification of the injector condenser, is made by the Worthington company. As in the injector type, the condenser proper is placed about 30 feet above the hot well and the water falling, through the action of gravity, creates the vacuum. There is no contracted throat to this condenser, however, and the water is sprayed into the head, where it becomes intimately mingled with the steam before discharging through the vertical pipe. Fig. 10 shows a section of the condenser head. An air cooler and a dry vacuum pump are employed, such as used with surface condensers, and any air that accumulates in the condenser head, where the steam is condensed by the spray of the water, is removed by the pump.

An interesting application of a jet condenser to a Parsons turbine is shown in Fig. 11.* The novelty of the scheme consists in substituting a centrifugal pump for the usual barometric column, enabling the condenser to be placed under the turbine. The exhaust steam is led through a pipe, *A*, and a gate valve, *B*, into the condensing chamber, *C;* and there, it is condensed by a jet and flows into the opening of a centrifugal pump, which is driven by a belt from the pulley on the extended shaft of the turbine.

There is a check, *D*, in the discharge from the pump, and this discharge is also sealed by the outgoing water. At the top of the condensing chamber is attached the usual dry air pump connection.

The turbine is started as follows: Valve *B* is closed and the turbine is allowed to exhaust into the atmosphere by the automatic exhaust valve, *E*. At the same time the dry air pump produces a vacuum in the chamber which induces a flow of water. When the turbine is up to speed the pump is going with sufficient velocity to keep the condensing chamber clear and the valve, *B*, which for convenience is motor-driven, is opened, the free exhaust valve, *E*, promptly closed, and the turbine is running condensing. During the winter months this plant gave a vacuum of from 27½ to 29 inches and the cost was only one-third the cost of a surface condensing plant

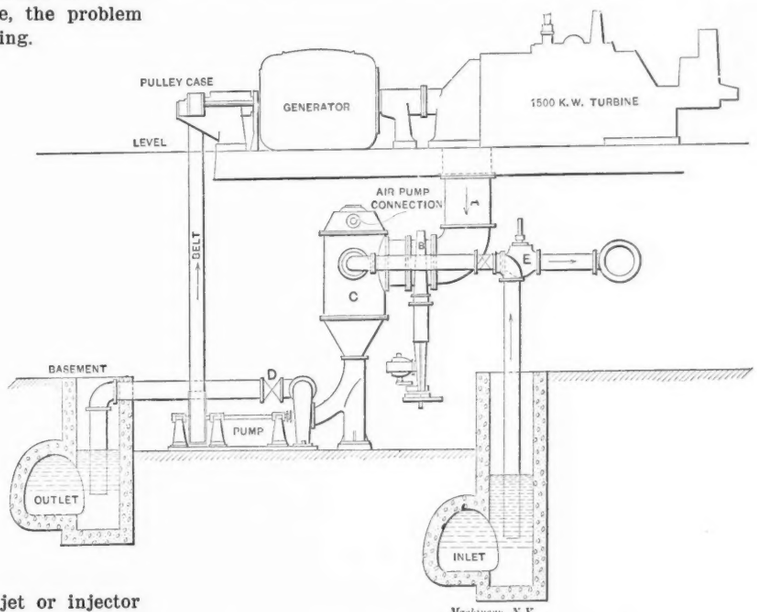


Fig. 11. Arrangement of Jet Condenser with Centrifugal Pump.

* * *

It is an interesting fact that the limitation of the speed of sewing machines, especially on thick goods, is not caused by the inertia of the reciprocating parts so much as by the heating of the needle. An ordinary steel needle will become so hot, working, say, at 3,000 stitches per minute, that it will turn blue and the temper will be drawn so as to make it useless. Strange to say, nickel-plating sewing machine needles reduces heating effect and considerably increases the speed with which cloth can be sewn. With the best modern lock-stitching machines, it is possible to sew at the rate of 3,000 or even 3,600 stitches per minute, using nickel-plated needles.

* Journal of Franklin Institute, May, 1905.

Copyright, 1905, by THE INDUSTRIAL PRESS.

Entered at the Post-Office in New York City as Second-Class Mail Matter.

MACHINERY

REGISTERED IN UNITED STATES PATENT OFFICE.

DESIGN—CONSTRUCTION—OPERATION.

PUBLISHED MONTHLY BY

THE INDUSTRIAL PRESS,

66-70 WEST BROADWAY, NEW YORK CITY.

LESTER G. FRENCH, Editor.
FRED E. ROGERS, Associate Editor.

The receipt of a subscription is acknowledged by sending the current issue. Remittances should be made to THE INDUSTRIAL PRESS, and not to the Editors. Money enclosed in letters is at the risk of the sender. Changes of address must reach us by the 15th to take effect on the following month; give old address as well as new. Single copies can be obtained through any newsdealer.

We solicit communications from practical men on subjects pertaining to machinery, for which the necessary illustrations will be made at our expense. All copy must reach us by the 5th of the month preceding publication.

NOVEMBER, 1905.

PAID CIRCULATION FOR OCTOBER, 1905,—22,066 COPIES.
Including 673 advertisers' copies, as part of their contracts.

MACHINERY is published in four editions. The practical work of the shop is thoroughly covered in the Shop Edition—$1.00 a year, comprising more than 430 reading pages. The Engineering Edition—$2.00 a year—contains all the matter in the Shop Edition and about 250 pages a year of additional matter, which includes a complete review of mechanical literature, and forty-eight 6 x 9 data sheets filled with condensed data on machine design, engineering practice and shop work. The Foreign Edition, $3.00 a year, comprises the same matter as the Engineering. RAILWAY MACHINERY, $2.00 a year, is a special edition, including a variety of matter for railway shop work—same size as Engineering and same number of data sheets.

The statement of paid circulation in the above paragraph for the October issue should read: "Paid circulation for *September*, 1905—22,397 copies," instead of August as it erroneously appeared.

* * *

CONCENTRATING POWER TO REDUCE SMOKE

It is announced that Pittsburg is to be smokeless—smokeless for Pittsburg, that is—through the enterprise and foresight of George Westinghouse and H. C. Frick. Two immense power plants are projected, to be located outside the city limits, from which power is to be distributed to Pittsburg and other nearby cities or towns. About 3,000 acres of coal land have been purchased, and instead of shipping the coal to Pittsburg and burning it in the furnaces of the various factories and large buildings, the coal will be burned at the mines and power transmitted to Pittsburg.

While such an enterprise is a step in the right direction, it is to be doubted whether it will ameliorate to any considerable extent the almost intolerable conditions that exist most of the time in the "smoky city." This would depend somewhat on the number of power users that could be induced to purchase power from the new company. It would not diminish the smoke pouring from the stacks of the great steel works, situated up the Monongahela valley, which the wind sweeps down to the city in great black clouds, and, together with the fog from the rivers, extends the darkness of the night into the late morning hours.

The proposition to establish a plant at the coal mines and transmit power to points where it is to be used is an old one, and has been talked of for many years as a means of making use of the immense culm heaps at Scranton and other mining centers. There may eventually be many such plants established, and when the time arrives, which geologists tell is not very distant, that anthracite coal is a luxury and we all have to burn bituminous coal, it may prove that the soft coal nuisance is not so bad a nuisance after all, because heating, lighting, and power may then all be accomplished through electrical transmission from central stations. In such stations combustion of fuel can be regulated so that the smoke is not very objectionable; and the stations can be located where such smoke as does come from their chimneys will not reach the thickly populated sections. Some coal, however, will always have to be burned on the spot where the heat or power is required and the territory now served by the anthracite mines is destined some time to become a soft coal district.

* * *

WHAT IS UNSKILLED LABOR?

One of the contributors of MACHINERY, H. L. Millar, Manchester, England, has written us a little note protesting against the word "unskilled" labor. He very properly insists that there is no such thing in industry as unskilled work. Every effort that requires training, however small, to accomplish productive results is necessarily skilled; and the difference between it and so-called skilled labor is only one of degree. He suggests that there should be some other term used as a substitute for "unskilled," since that distinction is erroneous. The point is well taken, and we have often pondered on the subject for the question has frequently arisen with us as to what skilled labor really is. The ability to cook an appetizing dinner and to serve it in good shape requires manual ability and judgment of a high order; yet it is not usual to speak of domestic labor as being skilled. On the other hand, it is commonly thought of a watchmaker that he is highly skilled in the use of tools; although he may be a very awkward person with any other tools but those pertaining to his trade, and the chances are that he is. A farm laborer usually has a knowledge of the use of tools and an understanding of machinery that will put to shame many artisans who regard him and his calling with contempt. In short, a good laborer must generally have trained all muscles of his body and his sense of perception in such a way that he is competent to handle almost any job that may come his way. But the skilled artisan has generally trained only one set of muscles and perceptive faculties, and is consequently a very one-sided sort of person, both muscularly and mentally.

* * *

"HEROIC METHODS OF DOING WORK."

One of our correspondents finds fault with the heroic method employed in fitting the valve in the case of the hydraulic press repair described in the August issue. In a veiled way he gives the editor a jab for commending the "get-there" mechanic who sometimes uses means that are not sanctioned by the generally-accepted methods of the chaft. We think that our correspondent has attached too much importance to the editorial note on the subject, or has gotten a wrong idea from reading same. It was by no means our intention to commend lax or butchering methods of doing mechanical work. On the other hand, no one has a higher appreciation of what nice work is and of methods that must be followed to accomplish same. The point made in the editorial, or the one we intended to make, was that in time of stress the able foreman or superintendent is the one who can "cut corners" and "wrest victory from the jaws of defeat." In other words, he must take chances and do things that his better judgment would not commend when following the ordinary routine. It might be likened to the man who jumps from the dock to the steamer when the gang-plank is drawn in; if he reaches the deck safely, he is able to go on his trip and fulfill his engagement; and if he misses it he gets a briny bath—and perhaps worse. And in general, it is the man who takes chances that succeeds in the larger affairs of life. The successful merchant, financier, manufacturer and politician is the man who sometimes takes a "long chance"—and wins out. when he does so it is usually because it is either taking the chance or being defeated. He might as well be defeated in one way as another, but by doing a certain stunt there is the possibility of securing that which he desires.

* * *

The *Patternmaker* for October makes its appearance in a new form, as *Wood Craft*. The paper has been enlarged to the so-called standard size, 9 x 12 inches, and its scope has also been enlarged to embrace woodworking. The first issue is a creditable undertaking, and contains much that should be of interest to the craft in general. The leading article is an illustrated description of the woodworking department of the Eastman Kodak Company at Kodak Park, near Rochester, N. Y. Robert I. Clegg is the editor of the new paper; the publishers are the Gardner Publishing Co., Caxton Building, Cleveland, Ohio.

ENGINEERING REVIEW.

CURRENT EVENTS, TECHNICAL AND MECHANICAL—LEADING ARTICLES OF THE MONTH.

New Orleans is now said to be the largest sulphur exporting port in the world, the product being obtained from beds in Calcasieu Parish, where a sulphur strata 700 feet in depth exists. Nearly 15,000,000 pounds of sulphur of 99.5 purity have been exported from this port for the first nine months of 1905. So pure is the sulphur that the custom authorities of France class it as a refined product, notwithstanding the fact that it has been through no refining process whatever.

It is rather difficult for a person inexperienced in the matter of low temperatures to realize that fine gun oil may congeal to such an extent as to render a firearm useless in time of emergency. A recent issue of *Graphite* quotes from a sporting paper, which states that at a temperature of 72 degrees below zero a hunter found that the oil on the lock and firing-pin was frozen solid and it was impossible to fire the weapon. The use of graphite did away with this difficulty, as it made no difference how cold the weather was, the lubricant was not affected. It is worthy of note in this connection, that at exceedingly low temperatures some of the smokeless powders are greatly deteriorated, and in some cases refuse to explode at all.

Consul General Guenther, of Frankfort, Germany, refers to the Cowper-Coles process of galvanizing, which has been mentioned several times before in these columns. An interesting statement was made regarding the galvanizing of copper by the Cowper-Coles process, which is that a copper rod can almost be entirely transformed into brass, while the temperature employed remains far below the melting point of both metals. The possibilities of the process are somewhat startling as this would seem to indicate that it may be possible to manufacture brass wire and rods directly from drawn copper without the intervention of the brass founder. An advantage of this process which has previously been alluded to is the evenness of the coating, it being so perfect in this respect that galvanized screws may be made to fit their nuts perfectly while ordinarily when galvanized they have to be recut or loosely fitted.

The limitation to the use of the ordinary projecting lantern such as is used for throwing pictures on a screen, is that all the slides must be transparent. A new lantern has been developed of the reflecting type which makes it possible to throw a greatly enlarged view of any subject upon a screen in its natural colors without previous preparation. Its value for the purpose of illustrated lectures can scarcely be over-estimated, especially in the case of biological work. In the case of lectures on mechanical subjects, it can be made to have a living interest that is impossible with the ordinary lantern. The works of a watch for example can be thrown upon the screen at almost any desired magnification while in motion, and the separate parts may be shown in a dissected state for the purpose of explanation. In the case of metallurgy it might be possible to throw a highly magnified view of metal specimens on the screen and in this way get the true effect that is largely lost when resorting to photographic views, especially when made on a transparent plate.

At the recent general meeting of the Association of German Engineers, held at Magdeburg, Dr. Eichberg, of Berlin, presented a paper dealing with the recent progress and future of electric train haulage. In the course of his observations the author stated that the Prussian railway ministry had come to the conclusion that, from an electro-technical point of view, neither the double overhead conductor nor the third-rail was suitable for the working of trains on large railway networks. A technically satisfactory solution, as had been clearly proved by the experiments at Spindlersfeld, near Berlin, was afforded only by the combination of a single aerial wire with high-pressure current and single-phase motor. The same system had been employed with success in the Stubbai Valley, Innsbruck. It practically presented no material difficulties. A pressure of 5,000 volts to 6,000 volts has been used on the Spindlersfeld line, and it would be possible to employ a pressure as high as 15,000 volts on mountain railways. It was intended to work the line at a pressure of 6,000 volts, the trains to be composed of motor cars and trailers in accordance with the traffic requirements. The proposal to build a high-speed electric railway between Dusseldorf and Cologne now forms the subject of negotiations between the local authorities of the two towns, with special reference to the discussion which has taken place with representatives of the Allgemeine Electricity Co. and the Siemens-Schuckert Works Co.

I remember a bicycle shop, says a writer in *Cycle Trader*, where a great many frames were built and where consequently a great deal of filing-up was done. The filing-up was done at piecework rates, and it was, of course, natural that the men on this particular job should try to get through as many frames in a day as possible. Naturally, also, they would get as many new files as they could, and equally naturally the manager would see that all files were worn out before being replaced with new ones. Each man, when started, was given a full set of files, and when he required a new one, the old one, either worn out or broken, had to be delivered into the stores before the new files were sent out. The storekeeper, after seeing that the files returned were really unfit for use, issued new ones and threw the old ones on the scrap-heap. Now, the scrap-heap was the common rubbish heap of the shop, to which all had access. What was more natural than that, when a man wanted a new file so as to cut quicker and get through more work, even if his old file was not nearly used out, he should go to the scrap-heap, select the oldest file of the class he wanted, throw away his half-used file and stalk up to the stores with the old file, which had probably done the same duty hundreds of times before, and return with a new file. And so things went on. The storekeeper only gave new files out against absolutely worthless ones, which he threw on the scrap-heap; the men always had new files when they wanted them, and everyone was contented. The scrap-heap collected quite a little stock of half-used files, the men made good piecework wages, and the proprietor paid for about twice as many files per week as were needed.

The development of the electrical industry has made a great demand for pure copper, inasmuch as a small percentage of impurity in copper considerably increases its resistance. The most practical way of producing pure copper is to deposit it by electrolysis on the cathode plate, but there are practical difficulties in making copper products by electro-deposition, especially copper wire, which, of course, is much in demand.

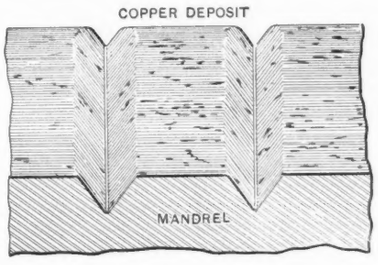


Fig. 1.

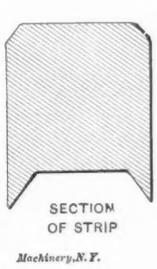


Fig. 2.

At the July meeting of the Faraday Society, Mr. Sherard Cowper-Coles read a paper describing his development of the centrifugal process of electro-deposition. He is able to produce copper wire at a cost of from $50.00 to $75.00 per ton less than by the usual methods. The cathode is made in the shape of a huge mandrel about six feet in length and seven feet in diameter. It is mounted with the axis vertical and is rotated in the copper plating bath at peripheral speed of about 1,000 feet per minute. In this way the rough "tree" or nodular growth,

which is a troublesome feature of electrolytic deposition ordinarily, is avoided; but the novelty of the improved process lies in the shape of the mandrel surface, and its effect on the copper plating deposited. A V-shaped spiral groove is cut in the periphery of the mandrel from one end to the other. Now when the copper is deposited on this cathode, the V-shaped groove causes cleavage planes to be formed in the metal as shown in Fig. 1. When the required thickness has been deposited, it is possible to strip the metal off in a continuous wire, having the approximate section shown in Fig. 2. With a mandrel of the diameter mentioned having the V-groove cut with a lead of 1-16 inch, nearly 5 miles of the strip will be deposited, and of course a greater length of wire, the length depending upon the reduction in diameter in the drawing dies.

BRITISH IMPROVEMENT IN HAND REAMERS.

Hand reamers of the fluted type as ordinarily constructed, says the *Mechanical Engineer*, are liable to act irregularly, and so cause the surface of the hole to be uneven. To overcome this difficulty, Chatwin & Slater, of Birmingham, have designed and patented the design shown in Figs. 1 and 2. As will be seen, the reamer increases in diameter, from the point or lower end which first enters the work, by a series of steps or shoulders, as *C D E*, the cutting portions being the narrow steps or shoulders themselves, while the portions between the steps or shoulders are made parallel, or nearly so, and serve only as guides for the cutting edges. The parallel portions have no cutting action. The number of steps or shoulders varies according to the amount of metal to be removed from

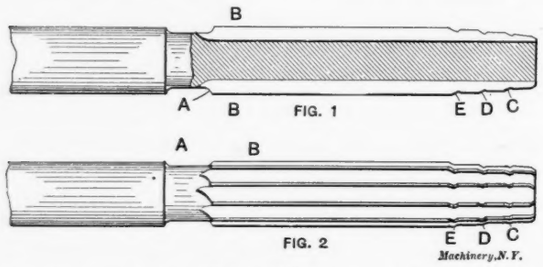


FIG. 1

FIG. 2

Machinery, N.Y.

the aperture or hole. The portions of the teeth, *B*, between any pair of the shoulders, such as the portions between *C* and *D* or *D* and *E*, are all ground practically parallel. The portions between *C* and *D* are ground to a diameter slightly less than the diameter of *D*, while the portions between *D* and *E* are ground to a diameter slightly less than that of the shoulder *E*. The portion between the first shoulder, *C*, and the bottom extremity of the reamer is ground to a diameter slightly less than that of the cutting edge formed at the shoulder. *C*, and it is made to taper down slightly toward its extremity. The reamer thus comprises a step-like series of cutting shoulders of progressive diameters with guiding and steadying surfaces between the steps, and with this arrangement the inventors have secured very effective and satisfactory results. Also the reamers may be readily sharpened by grinding the cutting shoulders without losing their size.

RAILWAY RATES IN ENGLAND.

Consul Mahin, Nottingham, Eng., in Consular Report No. 5.378.

Regular passenger fares on British railways are not absolutely uniform on all roads for similar distances. First-class fare ranges from 3 to 4 cents a mile, second-class is about 2½ cents, and third-class about 2 cents. Distance cuts little figure in the rates except where several lines compete for business between two important points—London and Liverpool, for instance—when the longer routes make their fares the same as those of the shorter route. First-class accommodations are better than those of the ordinary American day coaches, but hardly equal to those of our drawing-room or parlor cars. The first-class compartment is more expensively upholstered and more roomy than the second and third class. Usually it is arranged to seat only six persons; the other classes eight. The first-class is comparatively little used, and then chiefly by persons who want seclusion or who ride on passes or season tickets. The familiar remark that only royalty and Americans travel first-class in England is hardly pertinent now. My own observation is that Americans generally ride second or third-class. Both are sufficiently comfortable, and are seldom crowded, long trains being uniformly run, and Americans (excepting the uninitiated on their first visit) generally consider it a mere waste of money to buy first-class tickets.

The second and third-class compartments are both upholstered, contrary to the continental system, where third-class means a plain wooden seat. The casual glance sees no difference between the British second and third-class, but the second has a better quality of upholstering. Some of the railways have, however, dropped out the second-class, and use only first and third, the second being so little used. None of the railways traversing this consular district provide the second-class. Nearly everybody rides third-class. It is essentially as good as an ordinary American day coach, and therefore the fare of about two cents a mile is lower than the usual fare in the United States.

But the multiplicity of special excursion and week-end rates makes it possible to travel almost anywhere, at almost any time, for a fraction of the regular fare. These low rates usually apply to only the third-class, in this locality, but sometimes are extended to the first, when one may travel first-class at less than ordinary third-class fare—that is, for less than two cents a mile in a compartment nearly equal to the American Pullman or parlor car. For instance, the regular fare from Nottingham to Skegness, a seaside resort 73 miles distant, is $2.37 first-class and $1.50 third, one way. Round trip week-end (from Friday to Tuesday) tickets costs $1.94 first-class and $1.21 third, or one and one-third cents a mile first-class.

Every week one or more special attractions in London give occasion to offer low excursion rates, and in addition, every Saturday round trip reduced rates are given on one-half day up to six-day tickets. The regular third-class fare from Nottingham to London, 125 miles, is $2.49. The special round trip fares are $1.03 for half a day, $1.82 for one day, $2.31 for two days, $2.67 for five or six days, and $3.40 for eight days. Only the half-day tickets are limited to special excursion trains. The others are good on ordinary trains. Most local people who can arrange to return within the time limit go to London on these tickets. As would naturally be expected, people who do not intend to return often buy day or half-day tickets because they are cheaper than the regular one-way fare, and sell or give away the return coupon in London; but in spite of this the railways evidently find advantage in continuing such rates.

Similar reduced fares are constantly being given to both near-by stations and distant points on the islands, and for the round trip are less or little more, depending on limit of ticket, than the regular fare one way. In short, if the traveler can suit his convenience to the particular days of the week when reduced rates are given, and to the trains, of which there is often a choice of several, he need never pay more than half the schedule passenger tariff on English railways. The acme of cheap traveling in this country was reached this summer. Once or twice each week railroads give excursion rates from London and provincial towns to seaside resorts which range from five to nine miles for a penny (two cents). These are not on slow way trains, made up of obsolete cars, but on fast "expresses," some being non-stop and composed of new corridor cars. Taking account of all these reduced fares, it is probable that English railway travel is the cheapest in Europe; and, withal, the English railways and their service are inferior to none.

PATENTS IN JAPAN.

British Trade Journal.

There have been many cases of Japanese imitations of foreign inventions not patented in Japan, and in the industrial awakening, which may soon be expected in Japan, such instances are likely to increase in number rather than decrease. When such time comes, those who have been lax about taking out Japanese patents in the past may discover that their goods cannot be patented, because they have been introduced and offered for sale and have thus become "publicly known" in a way that deprives them of the right of patent.

The following notes on Japanese patent law will be of service to inventors who may desire to have their inventions patented in Japan. The Japanese Patent Office is the Patent Bureau of the Department of Agriculture and Commerce. According to an Imperial Ordinance, No. 234, dated December 4, 1903, the bureau is organized with one director-general, four secretaries, four technical officials, 15 examiners, 20 assistant examiners, 10 clerks, and three draftsmen. The supreme court judges patent suits only in cases mentioned in Article XXXV. of the Patent Law of Japan. This law, like that of the United States, is based on the examination system, and only those applications are entertained that are found to be patentable under the examination of patent officers.

There are three ordinary stages of examination, first and second examinations and trial. Those who may not be satisfied with the denial of a patent in the first examination can apply for a second examination within the 60 days, counting from the date when the notice of the denial was received. In case of denial in the second examination, a similar remedy is allowed for trial, which is the last resort in ordinary cases. The term for which a patent is granted is 15 years, counting from the day of registration.

The patent fee payable to the Patent Office is 10 yen annually, which is increased progressively by five yen after every three consecutive years; thus the total amount of patent fees for 15 years is 300 yen. Fees payable to the Patent Office for applications are five yen ($2.50), three yen ($1.50), and 12 yen ($6.00) for first examination, second examination and for trial, respectively. Any person wishing to make an application or claim with reference to a patent, or any patentee who has no residence in Japan, is required, according to Article VI. of the Patent Law, to appoint an agent who has a residence in Japan. Such agent is authorized by the same article to represent the principle in proceedings before the Patent Office and in civil actions and denunciations with regard to patents. Thus an agent representing foreign inventors in Japan ought to be a most trustworthy and responsible person, and the uncertainty of foreign inventors as to whether they can find such a reliable agent in Japan seems to be one of the chief reasons why they hesitate to patent inventions in that country. Trustworthy agents, however, are not entirely wanting there.

There are two other important facts to be remarked about Japanese patents. In the first place, the strictness of examination of applied inventions has been mitigated to a certain extent during the last few years, and secondly, the interpretation of the words "publicly known" under Article II. of the Patent Law has recently been changed. These two facts are of great importance to foreign applicants. As to the former, hitherto the patent officers in examining the applied invention, deviated from the practical point of view, dwelling much upon the theory of the invention. Consequently, the examination operated severely upon applied inventions, and many inventions which might have been patented under a more moderate and practical examination, were held to be not patentable. But things could not go on in this way, and recently the treatment of applications has been much more moderate, and they have come to be examined from a practical point of view. As to the second fact, hitherto the Patent Bureau strictly adhered to the view that an invention is not patentable if it was made known in some other country, even if it were not known in Japan; that is to say, the words "publicly known" were interpreted to mean a state of international publicity and many foreign inventions applying for patents in Japan were rejected on the ground that prior to the application in Japan they were known in the countries of origin through the official gazette of those countries. Recently the bureau has changed its views on this point, and interprets the word to mean only territorial publicity in Japan; that is to say, invention is not patentable there only when it has been made known in Japan prior to the application. Thus many inventions which were formerly held not to be patentable under the wide interpretation of the words "publicly known," have become patentable under the new interpretation. It will not be altogether useless to add a few words on the question as to what facts contribute to the existence of "public knowledge" of an invention under Article II. of the Japanese Patent Law. In case of a foreign invention that fact chiefly consists of the exposition, in the library of the Japanese Patent Bureau, which is open to the public, of the official patent gazette of the original country containing the description of the said invention. This is the reason why foreign inventors must apply for Japanese patents prior to such exposition of their inventions, and why it is most advisable to apply as soon as possible. The above remarks refer especially to patents. With regard to trade-marks, the term of exclusive use of a registered trade-mark is 20 years and the fee payable to the Patent Office after the registration is 30 yen ($15.00).

CONNECTING-ROD BOLTS FOR GAS AND OIL ENGINES.

Mechanical World, September 22, 1905.

In view of the report lately issued by the Vulcan Boiler and General Insurance Company on breakdowns of gas engines, especially with regard to the failure of connecting-rod bolts, it may be of some interest to investigate the stresses to which they are subjected, a point which is apt to be overlooked by designers of this class of machinery. The stresses are almost entirely due to the inertia of the reciprocating parts, and while this need not be considered in steam-engine design, the steam pressure imposing a much higher stress than that due to inertia, which it also opposes, in gas-engine work they amount to quite a serious matter, especially in the larger sizes.

The calculation for the maximum inertia stresses, which are the only ones that need be considered, can be easily made by the following formulas:

$$Q_1 = \frac{P v^2}{g r}\left(1 + \frac{r}{l}\right)$$

$$Q_2 = \frac{P v^2}{g r}\left(1 - \frac{r}{l}\right)$$

where

Q_1 = the accelerating force at the commencement of the outstroke.

Q_2 = the accelerating force at the commencement of the instroke.

P = the reciprocating weight in pounds.

v = the velocity of the crankpin center in feet per second.

r = the radius of the crank circle in feet = ½ stroke.

l = the length of the connecting rod.

g = gravity = 32.2.

Q_1 will put the rod in tension, as it has to pull the piston out. Q_2 will put the rod in compression, so need not be considered.

Assuming that the connecting rod equals 2½ times the stroke, and that the weight of the reciprocating parts is equal to a pressure of 4 pounds on the piston area, which is about the average for the usual construction of piston, the formula may be simplified to the following form:

$$Q_1 = 0.00064\ d^2\ s\ n^2;$$

where

d is the diameter of the cylinder.

s the stroke in feet.

n the number of revolutions per minute.

Taking an engine with 18 inches diameter cylinder, 24-inch stroke, and 160 revolutions per minute, the pull on the bolts becomes:

$$Q_1 = 0.00064 \times 324 \times 2 \times 25{,}600.$$

$$= 10{,}650 \text{ pounds} = 5{,}325 \text{ pounds on each bolt;}$$

which with a stress of 4,100 pounds per square inch would give bolts 1½ inch diameter.

If now we increase the speed 20 per cent, giving a piston speed of 768 feet per minute, which is by no means excessive in electric-lighting engines, the strains will be increased by the square of the difference in speed, viz.:

$$\frac{5{,}325 \times 120^2}{100^2} = 7{,}675 \text{ pounds,}$$

an increase of 44 per cent, requiring a bolt 1¾ inch in diameter. Of course, any increase in the weight of the piston due to scavenging or water-jacketing devices will increase the load on the bolts proportionately.

The formula given above can be easily modified to suit any design of engine by altering the constant in proportion to the variation of the reciprocating weight above or below that given and the proportionate length of the connecting rod.

The working stress of 4,000 pounds may appear low, but when it is considered that the strain comes on almost in the nature of a blow, and that the bolts are, or should be, in a state of initial tension with the brasses hard together, the margin of safety is by no means excessive.

While the above remarks apply more especially to the big end of the connecting rod, they are of equal importance in the design of the small end, which is generally wretchedly designed in its means of adjustment, the bearing surface between the adjusting brass and the tightening screw or other device being ridiculously small, the result being that this end is the first to show signs of hammering. Some makers appreciate this, and fit a wedge piece with large bearing surface similar to that used in steam-engine practice; but these are the exceptions, especially in the smaller sizes of engines.

THE ESPIONAGE AND THE HONOR SYSTEMS.

New York Commercial.

Modern business, with its clocklike system, its minutely subdivided work, its elaborated scheme of supervision, its almost Bertillon-like methods of keeping tab upon its workers, is becoming more and more a complicated piece of machinery wherein the heart of man and the finer sensibilities of the worker have less and less place. Man is the instrument used to polish, feed, engineer and keep in motion this tremendous commercial engine that it may puff, steam and grind out wealth, more wealth. The industrial machine that grinds is so tremendous, the wealth that is ground so huge, that man, the little implement in the vast mechanism, seems but an atom.

The old personal relation between employer and worker is lost. In the new scheme of things the employer sees the worker only as a mass, only rarely or never as a unit; and then only at infrequent intervals, when he, by chance, strolls through the establishment that bears his name. The worker knows the employer only through the cunningly devised apparatus that runs the industrial machine—surely an impersonal relationship. This apparatus that to him represents his employer is a series of cunning devices that act like many eyes to spy upon him in his work, to make sure that no effort is lacking on his part to help along the grinding of the mill of gold.

These devices are time-clocks, timekeepers, time cards, floor walkers, overseers of every description and sometimes a side line of inspection on the part of his fellow worker. Personal honor, personal integrity, nobleness, disinterestedness of action as business-making elements are not looked for or taken much account of. The great factory, the vast corporation, the huge commercial house, mechanically controls in the gross, as it were, the moral principle of their working force. No man can take a bundle away from the house or factory without its being O. K.'d by a superior officer, even if the parcel contains only a pair of overshoes that he does not want to wear home.

Put a man or a boy who has been brought up on an honor system, such as ruled at Dr. Arnold's Rugby school or such as rules in many a classroom in America—put such a man or boy under such management and picture his feelings for the first day or so. He will imagine that all this espionage, this policeman-like method of insuring orderly work, is directed toward him alone. At first he will inwardly resent it as an imputation against his honesty. Later he will recognize its impersonal character and wonder if the moral sense of man has declined so that any such system is necessary.

He then sees that knots of workers here and there form informal unions to "beat the machine." The time-clock can be manipulated. One man can punch another in. He can also punch another out. When times are dull craps can be played behind the scenes and a little private pool on the races can be made, and all this against the rules.

Thefts even can occur, notwithstanding the strictest kind of espionage known in modern business. One of the houses of the most rigid discipline has yearly a number of thefts made by its employes. These cases rarely reach the newspapers, for the policy of the house is against publicity. The thief is found out, gives up his spoils, and if he makes a clean breast of it is simply dropped from the rolls. While under inquisition he has a chance to implicate a coworker, and he sometimes does this to pay off old grudges against an innocent man. Strange as it may seem, the firm sometimes believes a thief when found out, while before his crime his word would not have been of equal value.

In view of all this M. M. Atwater has been moved to ask this question and to answer it in part: Would not the firm that systematically could put a reasoned faith in the honor of the entire corps of workers be benefited thereby?

Women, as a rule, use the honor system with their help, relying upon their intuitive reading of character, and they are seldom mistaken. The schools and colleges that use the system of student government find that it works to a charm in turning out young men who can stand upon their own feet morally. If large bodies of youths can be so dealt with, why is it not possible, by putting faith in the moral sense of men in business, thus to stir it into action?

Put responsibility upon a worker and show trust in him, and he will be worthy of it—nine times out of ten. Kick him and knock him, you toughen his hide, you make him more insensible to affronts, but do you thereby increase his efficiency? Do you not take from him his self-respect, and sometimes his moral goodness?

The wise employer of the future will take the worker when he comes to him with ideals and aspirations that in later years seem a lost hermitage to be mourned, will take him, raw and green as he is, and by a courteous, considerate treatment will develop the morality of the boy, side by side with his business qualities. He will do this, not for altruistic reasons entirely, but because of the monetary returns to the house. He will know that if you treat the immature boy as a possible shirk, as a possible thief, as a possible moral delinquent, you put him in a fair way of becoming this possible thing.

A NEGLECTED POINT IN STEAM CONDENSERS.

William Booth in Electrical Review (London), September 15, 1905.

It is quite common, when calculations are made relative to the volume of air to be dealt with in a condenser, to neglect the law which governs the mixture of gases with saturated vapors. Thus, when the temperature of a condenser is, say, 104 degrees F., the pressure proper to that temperature is 1.06 pound per square inch, which represents a "vacuum" of $14.7 - 1.06 = 13.64$ pounds by gage. Supposing, however, that the actual gage reading is only 13 pounds, the additional 0.64 pound is held to be due to air, and the actual absolute condenser pressure is $1.06 + 0.64 = 1.7$ pound. It is very usual to assume that the air in the condenser is present at this pressure of 1.7 pound, neglecting the presence of water vapor.

Thus if air weighs 1 pound for each 13 cubic feet, its density in the condenser will be assumed to be such that the number of cubic feet per pound will be $13 \times \frac{14.7}{2.7} = 71$ nearly, and the air pump would require to generate nearly 5 cubic feet of volume for each cubic foot of air leakage. This view tends to the production of insufficient air pumps.

One of the laws of gaseous mixture is that "the pressure of the whole of a gaseous mass is the sum of the pressures of all its parts." A second law is that "the presence of a foreign gaseous substance in contact with the surface of a solid or liquid does not affect the density of the vapor of that solid or liquid," if there is no chemical combination between the two substances.

Rankine selects as an example in his work on the steam engine, steam at 212 F., 1 cubic foot of which weighs 0.03797 pound and exerts a pressure of one atmosphere. The pressure of one atmosphere is exerted by 0.080728 pound of air at 32 degrees F. If this weight of air be heated to 212 degrees F. it will occupy a cubic foot at the pressure of 1.365 atmosphere. Then if 0.080728 pound of air at 212 degrees be placed in a vessel of 1 cubic foot capacity with 0.03797 pound of steam also at 212 degrees F., their united pressure will be 2.365 atmospheres. Molecular equilibrium demands that each cubic foot of space in contact with water at 212 degrees shall contain 0.03797 pound of water vapor. Alone the vapor exerts

a pressure of one atmosphere. With air added the pressure is increased, yet the boiling point remains the same.

This latter fact is what is ignored, because engineers are so accustomed to consider the temperature, density and pressure above a mass of water to be three invariables. The pressure is invariable where air is not present. It is not, however, a fixed quantity when another vapor is present. Rankine, in further illustration of this law, assumes water at 50 degrees F., and a cubic foot of space containing the invariable 0.00058 pound of water vapor proper to that pressure, "whether and to what amount so ever air, or any other gaseous substance not chemically attracting the water, is contained in the same space."

This is the law discovered by Dalton and Gay-Lussac. In other words, any amount of gas may be pumped into a space filled with the vapor of water, without decreasing the weight of aqueous vapor previously existing in that space by virtue of the temperature of the water giving off that vapor from its surface.

Recognizing the law as correct, the density of the gas present with the vapor may be calculated as follows: $P =$ the pressure of the mixture, $p =$ the pressure of the vapor proper to the temperature, such, for example, as one atmosphere for water at 212 degrees F.

Then the density of the air or other mixed gas is reduced below that proper to the pressure P in the ratio $\frac{P-p}{P}$. Thus, to take our first example, $P = 1.7$ and $p = 1.06$. Then the density of the air $\frac{1.7-1.06}{1.7} = 0.38$ nearly. The density of air at 104 degrees and 1.7 pound pressure in pounds per cubic foot is

$$0.080728 \times \left(\frac{461+32}{461+104}\right) \times \frac{1.7}{14.7} = 0.0081$$

or, say, one-tenth of normal atmospheric density. But under the conditions named the density of air in a "vacuum" of 1.7 pound pressure above water at 104 degrees F. is only $0.38 \times 0.0081 = 0.0031$ nearly, or practically one-twenty-seventh of normal atmospheric density. Thus there is much less air in the condenser than would be present if the air really had the density corresponding to the pressure. It is, in fact, necessary not to neglect the law of mixed vapors. The pressure in a condenser cannot be greater than that corresponding with the temperature of saturated steam. Any additional pressure, *i. e.*, deficiency of "vacuum," must be caused by air. The presence of this air does not mean that some of the water vapor is pushed back into the condition of water, for the conditions of molecular equilibrium demand the presence of a fixed quantity of vapor per cubic foot of space proper to the temperature. A consideration of the foregoing law will help to explain why it is that the pressure gage on a boiler will sometimes fall so very rapidly when the engine is started. The steam space of the boiler has been more or less full of air while the boiler has been cold. When heated, the pressure gage reads the combined pressures of the steam and of the air, and this will be considerably above the pressure proper to the temperature of the boiler. As soon as the air is blown out, the steam pressure shows the actual temperature of the boiler.

Thus, a boiler is closed up tight, and it is then full of air. Feed is admitted, and the air is compressed, say, to 20 pounds absolute. When the boiler temperature is 300 degrees F. the air pressure will be 31 pounds, and the steam pressure will be 71 pounds. The gage pressure will be $(71 + 31) - 14.7 \times 2 =$ say, 72 pounds, and when the air is blown out, the gage will drop back to $71 - 14.7 =$ say, 56 pounds, being the gage pressure proper to 300 degrees F. of temperature, whereas the supposed steam pressure of 72 pounds is that proper to a temperature of about 315 degrees F.

In fact, it is possible that by an air trap, a boiler might be 15 degrees F. below its proper temperature, while the gage was showing full pressure. The fall in pressure will not be noticed when a boiler under these circumstances is put upon the main steam pipe, for its pressure will be maintained either by back flow from the other boilers to which, for a time, it will act as a condenser, or, if the stop valve can open only by pressure from below, there will be no flow of steam until the boiler becomes fully heated. This will explain why the putting in parallel of a boiler at full gage pressure may not have any effect in helping the work for a few minutes, but may cause the vacuum to drop considerably. Drop in steam pressure when the stop valve is opened has usually been explained to be due simply to stirring up of cooler water in the lower part of a boiler. This effect is not possible in under-fired boilers, and the explanation is simply as above.

To return to the condenser, however, if the volume, or rather weight, of air in this is not so great as the pressure would signify, but for the law of mixtures the same proportion of such weight will be abstracted by each stroke of the pump. The fact that air adds so greatly to the pressure in a condenser should, however, spur on the engineer to effect its complete elimination. The degree of approximation of the vacuum gage to the condenser thermometer is a measure of the degree of elimination of air. A good vacuum gage would be one with two pointers, which should coincide, if properly made to do so, when no air is present. The lag of the vacuum pointer will show the degree of preventible imperfection.

ZINC CASTING FOR ORNAMENTAL WORK.

Walter J. May, in Mechanical World, September 29, 1905.

For a good many purposes of inside decoration a hard-metal casting which can be bronzed or otherwise finished is a desideratum, and zinc, when alloyed with some copper, will give what is needed—that is, as a general rule; but, of course, brass or bronze is best. Each alloy has its drawbacks, however, and while brass and bronze have to be cast at from 1,700 to 1,900 degrees F., according to their composition, zinc alloyed with a small amount of copper casts well at about 800 to 900 degrees F., which is a great consideration at times, particularly when plaster molds are used. Even with fine-sand molds a metal which pours at comparatively low heat is an advantage. And assuming that decorative effect, and not value, is the object aimed at, then the most workable metal should be used.

The making of the alloy requires a brass furnace in which to melt the copper, and skill is required in alloying; but afterwards the metal can be melted as easily as plumber's solder. For the simplest form of alloy, only enough copper to break down the grain of the zinc is required, and this works out at about 92 pounds of zinc and 8 pounds of copper per 100 pounds of metal mixed. Usually, however, 90 pounds of zinc and 10 pounds of copper would be best, owing to impurities in both metals, the alloy thus made being hard and facing-up in the lathe with a good smooth surface, having a somewhat steely appearance.

Where an extra white metal is needed, an alloy composed of 88 pounds zinc, 9 pounds copper, 1½ pound tin, and 1½ pound aluminum, may very well be used, this finishing up bright and producing a white casting, the cost of the alloy being rather more. This can be cast to have an appearance almost like frosted silver, a color which is kept if the castings are coated with clear lacquer before they dull off.

In making the simple zinc-copper alloy, the copper is first melted in a clean crucible of sufficient size, and to this the zinc, in as hot a state as possible, is added piece by piece until the violent reaction ceases. Then the bulk of the zinc is added, stirring well, and then pouring into slabs or bars, afterward breaking up and remelting to insure a thorough alloy, and then pouring into conveniently-sized ingots for use when needed. With the quadruple alloy, the copper is first melted, then the aluminum and tin successively added, and afterwards the zinc, the latter entering more quietly than with the simple alloy. In the melting for pouring into the molds the metal can be melted in an iron metal-pot well coated with plumbago, care being taken that the metal is made fluid without being unnecessarily hot, and any not at once required should be poured into a bar for future use. Burning the metal should always be avoided, and equally should keeping it in a melted state for any length of time, as this latter will cause oxidation and a change of content which leads to much disappointment with the castings.

In molding in sand it is usually preferable to use a fine kind, like the Charlton brass sand, first drying and then passing through a fine sieve, and adding some pea or bean flour to the facing sand before damping down for use. It is advisable to use the sand as dry as possible consistent with good molding, and in some cases the molds may be surface-dried with advantage. Where color is not an object in the castings, the molds may be very lightly dusted with plumbago shaken through a fine cloth; but where a clean white surface is needed, this material cannot, of course, be used. A very slight dusting with steatite would be permissible, but the amount applied must be very slight. New sand, of course, will be used, but in practice, if the molds are faced with, say ½ inch of new sand and backed with floor or ordinary sand, this will answer all purposes, while it prevents undue waste of sand. Free venting to permit of the rapid escape of air and generated gases is necessary, otherwise the castings will often run up faint; but the alloys in question do not set with sufficient rapidity to prevent the formation of good castings.

Where metal molds are used, the alloys in question present only the ordinary difficulties; but they should be brushed out with plumbago or steatite to keep a good face on them, and to prevent the zinc from eating into the iron.

Plaster molds should be composed of about three parts plaster and one part finely powdered Bath brick, or some other fine sand, the actual proportions only being determinable in practice, owing to variations in the quality of the plaster. Plaster molds should be thoroughly dried before the metal is poured into them, and it is a good plan to run a little metal through such molds to insure their filling. There is necessarily some experience required before plaster molds can be satisfactorily dealt with; but this is one of the incidental experiences of molding practice.

With ordinary care the castings leave the molds very cleanly. If they are to be bronzed or otherwise treated, they should be well brushed with a fine wire brush after any tooling has been done to them. In turning, shaping, or other form of machining, keen tools should be employed, all work being done without lubricants, as such are necessary. The metal should also be polished, if necessary, instead of being burnished, as the burnisher does not work well on zinc as a rule. Where runners have to be removed, they should be sawn off with a fine hacksaw, the teeth of which allow plenty of clearance for the body of the saw, such makes as the "Star" giving plenty of clearance when new. Filing should be done with not coarser than second-cut files, and these should be kept well chalked to prevent them pinning up. Although the alloys dealt with, file with freedom when the files are kept dry and free from grease, yet when greasy they pin up very badly, and when in this condition files will not do good work. Filed work should be finished with files of successive fineness, this working up a better face than using emery cloths; but the final polish may be given with the finest emery cloth, if so preferred.

Bronzing and the like can be done in the usual way, but plating on zinc is more difficult than on brass, and should only be attempted by skilled platers.

NOTES ON THE DESIGN OF LARGE GAS ENGINES WITH SPECIAL REFERENCE TO RAILWAY WORK.

From Paper of Arthur West, read before the Philadelphia Convention of the American Street Railway Association.

One of the most important considerations in the design of large gas engines is the arrangement of the cylinders. In a single-cylinder, single-acting, 4-cycle engine an explosion takes place once in every two revolutions. In order, therefore, to get the same rotative effect as with a double-acting steam cylinder, it is necessary to work four single-acting cylinders on the shaft or two double-acting gas cylinders tandem on one crank pin. With this arrangement four explosions are obtained in two revolutions, or an explosion every 180 degrees of crank angle. In case of a misfire or premature ignition due to bad gas, the crank can only move one-half a turn before another explosion takes place. In a single-cylinder, single-acting engine the crank must move two whole turns before the next explosion, while with two single-acting cylinders opposed to each other or one double-acting cylinder, the crank may be required to move one and one-half turns before the next explosion. The relative evil effects of a premature or misfire are, therefore, in the following ratios:

Two double-acting cylinders	1
Two single-acting cylinders, opposed type	3
One double-acting cylinder	3
One single-acting cylinder	4

Gas engines and producers to be commercially successful must be designed to be run with the same class of help as is employed on Corliss engines and boilers. This being the case, misfires and prematures are liable to occur occasionally, and the designer must minimize their possibilities for evil. These considerations, as well as the capacity for caring for heavily swinging railway loads, have caused our adoption of tandem double-acting cylinders for railway work.

It is sometimes argued that cylinders so arranged are inaccessible. If, as is the practice of the Westinghouse Company, ample space is arranged between the cylinders, and if the inlet and exhaust valves are not located in the heads, but in the cylinder body and entirely above the floor level, such a gas engine is as accessible as a tandem-compound Corliss engine or as a Corliss engine driving an air compressor.

The speed of a gas engine must be adapted to the kind of generator to which it is to be directly connected and will usually somewhat exceed that of a Corliss engine of the same cylinder dimensions. The speed regulation adopted for large Westinghouse gas engines is especially suitable for generator driving, in that no conditions of changeable load or variable friction of valve gear affect the regulator. Our gas engine regulator governs the speed by means of a relay cylinder, and therefore produces results similar in type to those obtained with the relay governor used by the Westinghouse Machine Company on steam turbines. The advantage of such a relay governor with the gas engine is that the varying friction of valves with different qualities of gas does not affect the sensitiveness of the governor. Without a relay cylinder, the only way in which this result can be accomplished on large gas engines is by some form of a drop cut-off controlling the gas. This is objectionable on a gas engine, as any slight change in the speed of the dash pot very seriously affects the mixture of gas and air, with corresponding bad effect upon the regulation. Such small changes in speed of dash pots are frequent in a Corliss engine, where they cause no bad results. The Westinghouse arrangement employs no releasing gear of any kind, but secures all the advantages of regulation without its use.

The question is frequently asked as to whether large gas engines will drive alternating-current generators successfully in electrical synchronism or "parallel." This has been done for several years past in Germany with entire success, and it has also been done in a number of instances very successfully by our company. It is sufficient for our purpose to observe here that the cyclic variation—*i. e.*, the degree of departure from absolutely uniform rotation—is sufficiently small to conform with the design of generators now built for steam driving.

The European designer of gas engines has allowed himself an amount of complication in valve gear which would not be permissible under American operating conditions. The successful American machine must be as nearly "foolproof" as is the large Corliss engine. If it is not, it will fail to be a success from the purchaser's point of view—no matter what thermal efficiency may be claimed by the builders—as a consequence of such complication as the European engineers have been prone to adopt. In the designing of valve gears for large gas engines, wide range of quality of gases must be considered. In this respect the design of the gas engine is very different from that of a steam engine, inasmuch as the steam used has practically constant characteristics, differing only in such minor points as pressure and superheat. With the different kinds of gas to be met with, however, the proportions of air and gas, and sometimes of compression, are radically different, and no gear can hope to be a universal success which does not provide for meeting the widely varying conditions to be encountered in the market.

We are frequently asked, "What is the overload capacity of

your gas engine?" A clear understanding in this direction is very desirable, from the point of view both of the buyer and the seller. A gas engine and producer is thermally very much more efficient than a steam engine and boiler. It is, perhaps, not amiss to say that, with a well-designed producer and gas-engine plant, a horse-power can be delivered with one-half the cost of fuel that is possible with a well-designed steam-engine plant. The power of the gas engine, however, is limited by the total volume of explosive mixture which can be drawn into the cylinders during the suction stroke, compressed and finally ignited. This condition sets a limit which does not allow of a large temporary increase of the power, such as is obtained with the Westinghouse steam turbine by the automatic operation of the secondary admission valve. Such overload capacity is, of course, convenient for the purchaser, but it is unobtainable on a gas engine, unless the engine is largely underrated, and the purchaser should consider that this is one of the prices that he pays for the enormously increased output obtained with the gas engine per pound of coal. The overload capacity is, therefore, simply the amount which the builder rates his machine below its ultimate capacity. It has been our practice to rate our gas engines in such a way that they would have a safe overload capacity of 10 per cent. Our machines are ordinarily good for somewhat more than this, but conservative engineering requires that there be a margin of power in order that overloads may not materially reduce the speed. The above remarks on overload furnish a general guide which may be of service in selecting suitable generator capacity for a gas engine. For ordinary cases the overload capacity of the generator and that of the gas engine should be about equal, although the gas engine will indefinitely carry its overload, while the generator will not, in all cases, unless it is bought with that understanding.

The mechanical efficiency of a large gas engine is very much greater with a 4-stroke cycle than with a 2-stroke cycle, this being one of the arguments against the 2-cycle engine. It is no uncommon thing to see 2-cycle engines which do not realize as brake horse power more than 60 per cent of the work actually done by the cumbustion in the cylinders. The efficiency of a 4-cycle engine varies considerably, but it may be said in a general way that a well-designed engine will deliver about 85 per cent of the gas indicated horse power in the form of brake horse power. This 15 per cent of power lost is not exclusively composed of frictional resistance of journals, cross-head slides, etc., as is the case in a steam engine. The 4-cycle engine has, of course, to draw in its own mixture of air and gas and compress the same, and its functions, therefore, combine those of a pump, a compressor and a motor. It is the pumping and compressing work which causes the mechanical efficiency of the gas engine to be somewhat lower than that of a steam engine. The actual friction of the working parts need be no greater than with a well-constructed Corliss engine, viz., 90-95 per cent. In order to keep down the friction and increase the reliability of the machine, it is the practice of the Westinghouse Company to design large gas engines with provisions for attaching a continuous return oiling system. The large amount of oil put through the journals increases the safety, requires less attendance and keying up, and washes out dust if the engine is required to operate in an atmosphere which is not clean.

There is an impression rather prevalent that a gas engine is uncertain and hard to start. A properly designed engine,

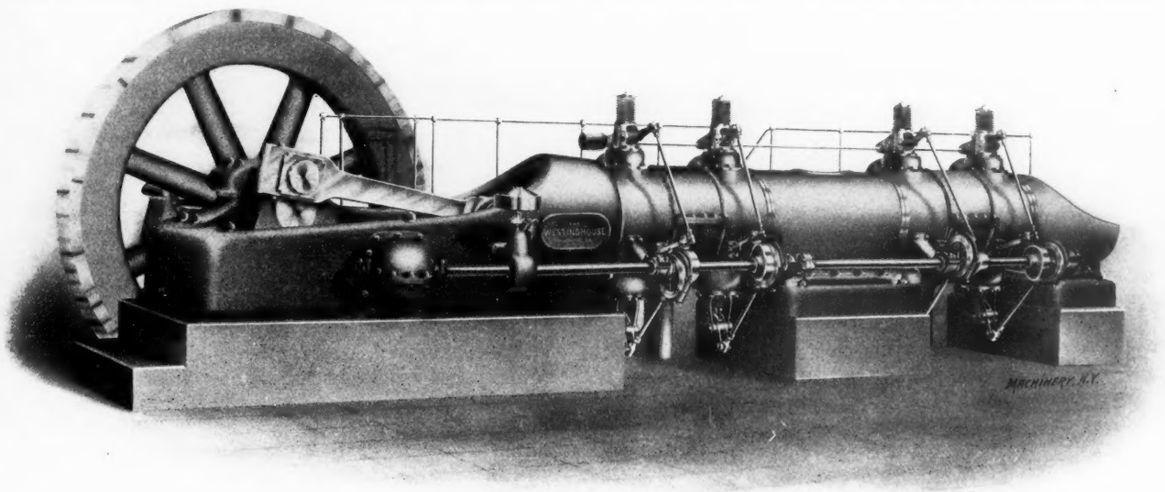


Fig. 1. One thousand H.P. Westinghouse Gas Engine for Electric Railway Service.

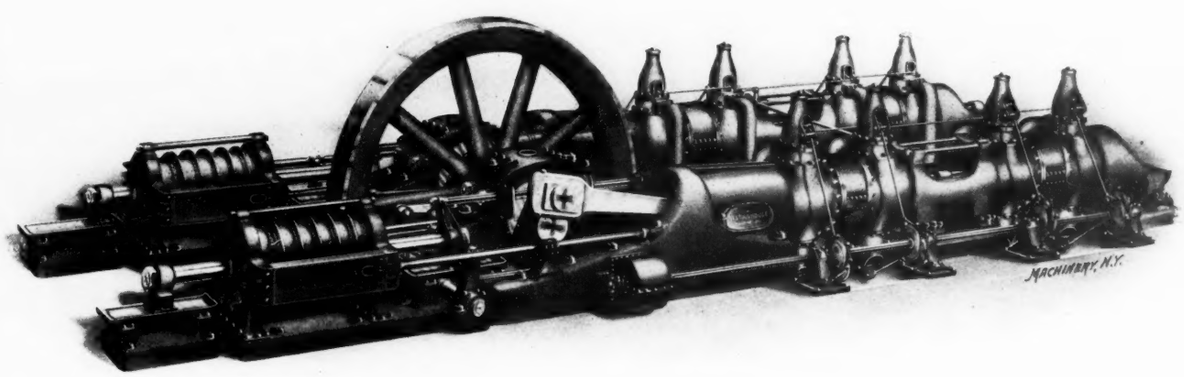


Fig. 2. Twin Tandem Blast Furnace Gas Blowing Engine.

supplied with fairly decent gas, can be started as easily as a steam engine. Large Westinghouse horizontal gas engines are started by means of compressed air, the only operations required being: (1) open the main gas valve; (2) close the igniter circuit; (3) open one compressed air valve, similar in construction to an engine throttle. The compressed air puts the engine in motion, which draws the charge into the cylinder and compresses the same, after which the first explosion takes place. Air is shut off and the engine is in full operation.

With certain kinds of gas, inspection of the interior parts of the cylinders is often desirable at regular intervals of, say, a couple of months. This is especially the case with blast furnace gas, and also with producer gas made from certain kinds of fuel. We have taken particular pains to arrange our cylinders so that no parts of the valve gear or valves are below the floor. The inlet valves being located directly on the top of the cylinder, easy access can be had to either end of either cylinder by removing the inlet bonnets. The exhaust valves are also a part of the engine which need occasional attention for regrinding. Especial care has been taken to render these quite easily removable. The cylinders are, therefore, directly accessible from the top through the inlet openings and from the bottom through the exhaust openings. The fact that all the valve parts are entirely above the floor line renders these operations comparatively easy.

The general type of engine commented on above is shown in the two accompanying illustrations. The first shows the type of two engines being built by the Westinghouse Machine Company for the Union Traction Company of Kansas, Independence, Kan., one being of 500-brake horse power and one of 1,000-brake horse power. The second photograph shows one of two twin-tandem furnace gas blowing engines now under construction for the Edgar Thomson plant of the Carnegie Steel Company.

* * *

MACHINE TOOL BUILDERS' CONVENTION.

The annual convention of the National Machine Tool Builders' Association was held at the Hoffman House, New York, on Monday and Tuesday, October 16 and 17. The Association now has a membership of 51, and there were over 40 present at these meetings. The following new members joined the association: E. Rivett, the Rivett Lathe Mfg. Co., Brighton, Mass.; F. W. Hoefer, the Hoefer Mfg. Co., Freeport, Ill.; W. A. Chase, National Acme Mfg. Co., Cleveland, O.; Israel H. Johnson, Jr., I. H. Johnson, Jr., & Co., Philadelphia, Pa.; George J. Burns, the Chandler Planer Co., Ayer, Mass.; Clarence Whitney, the Whitney Mfg. Co., Hartford, Conn.

The meetings of the association are held in executive session, and the business transacted is generally not for public discussion. It is stated, however, that the subject of general trade conditions and prices formed the basis of the proceedings, as is usually the case at these meetings, while there were no reports of importance from committees, relating to the changes in design or to uniform proportions for machine tools, as have sometimes been submitted in the past. There was, however, a report of delegates to the National Reciprocity Convention held last August at Chicago. It was the sense of this convention that there should be an immediate reciprocal concession to other nations to relieve the strained situation now existing, owing to our failure to carry into effect the reciprocal trade provisions of the Dingley law.

There has been a strong feeling among machinery dealers that there should be an advance in commissions by machine tool builders, and many of the latter class of manufacturers, while combating this idea, have been in favor of an advance in prices upon machine tools. The whole subject was threshed out at the several meetings, with the result that no action was taken by the convention as a whole, looking to any change either in commissions or in prices of tools. There are several standing committees, however, which have in charge the trade interests of the manufacturers of the different classes of machine tools and they may act upon the question of prices at their discretion. It is understood that the lathe committee have decided upon no advance at present, while the milling machine committee are in favor of an advance.

The officers for the ensuing year are the following: President, E. M. Woodward, the Woodward & Powell Planer Co., Worcester, Mass.; first vice-president, William Lodge, the Lodge & Shipley Machine Tool Co., Cincinnati, Ohio; second vice-president, W. P. Davis, Rochester, N. Y.; treasurer, F. E. Reed, Worcester, Mass.; secretary, P. E. Montanus, Springfield Machine Tool Co., Springfield, Ohio.

On Monday evening a banquet was held at the Hoffman House by the association, where members and their guests listened to addresses by Charles A. Moore, of Manning, Maxwell & Moore, New York, and Fred J. Miller, editor of the *American Machinist*.

Mr. Moore spoke upon the possibilities of foreign trade for American machine tool builders, and talked interestingly, giving the results of his experience and observation, extending over the many years that he has been so intimately associated with the machine tool trade. He contended that American tools were well thought of abroad, but that our merchants had not been as progressive nor as aggressive in disposing of their products as had our new competitors, the Germans, or even our old competitors, the Englishmen. Frequently the American manufacturer failed to secure foreign orders because he did not understand the conditions to be met, and even when he was successful he ran across disadvantageous conditions, as for example, the lack of banking facilities, except through London exchange.

Attention was called to several practical aspects of the case, as, for example, the necessity for careful boxing and shipping; since, when a foreign buyer receives a tool that has been damaged in transportation, there is a long delay before the damaged parts can be replaced, and the matter finally adjusted.

Mr. Moore dwelt at length upon tariff and reciprocity, and while expressing himself as favorable to any revision of the tariff law which would affect a considerable number of industries favorably, he advocated the present policy of protection under which the country has thrived, and cautioned his hearers against any proposition looking toward a sudden or radical change.

Mr. Miller read a paper upon government manufacturing in which he outlined some of the reasons why the manufacturing establishments controlled by the government operate at a disadvantage, as compared with independent concerns.

He advocated a cabinet department of manufacturers having a secretary of manufacture at its head who should be a practical manufacturer accustomed to organizing industrial establishments. The business of manufacturing by the government should then be removed from the control of the army and navy, and placed under the care of this cabinet officer. In other ways such a department would prove beneficial in bringing together the various testing departments or bureaus, and thus making standard the method used by the different ones. The manufacturing operations of the government should be carried on not by sailors and soldiers, but by manufacturers.

MACHINERY'S Annual Outing.

The entertainment of the Association and its guests by MACHINERY has assumed the form of an annual outing which will be given by this paper every year about this time. On the above occasion about three hundred and fifty manufacturers and dealers accepted MACHINERY'S hospitality on a trip around Manhattan Island on a steamer chartered for that purpose, leaving the foot of West Twenty-second Street at 1:15, Tuesday afternoon. After luncheon was served the entire party visited the Brooklyn Navy Yard; an hour was allowed for inspection of the interesting work in progress at this important naval station. The steamer then proceeded up the East River through the Harlem River and Spuyten Duyval Creek to the Hudson, and returned to the Twenty-second Street pier at about six o'clock. The sail through the Harlem River, which separates Manhattan Island from the main land, was particularly interesting. In order to follow this course it was necessary for twelve draw-bridges spanning the river to be opened to enable the steamer to pass, but there was no serious delay at any point.

This outing was the means of bringing together the largest and most representative assemblage of machine tool builders that has ever gathered at a single meeting place, and undoubtedly accomplished its principal object—the promotion of acquaintance and good fellowship among the trade.

TURRET TOOLS FOR DRILLING HOLES IN CAPSTAN-HEAD SCREWS AND NUTS.

C. V. RAPER.

There are a great number of manufacturers besides those producing scientific instruments and the like, who possibly have occasion to, or do use, both capstan-head screws and nuts, and probably in the case of the ordinary machine tool maker the nut is more in demand. But the employment of these good looking and compact adjuncts to machine construction is probably militated against, to a great extent by the fiend "Cost," which looms up at every point of the design of a machine. Speaking metaphorically, the tools to be discussed are the result of an attempt to exorcise the demon of expense and bring into more general use the capstan-head screw and nut.

The tool shown in side elevation and part section at Fig. 2 and in end elevation at Fig. 1 embodies and shows the main principle applied by myself to tools of this nature as used in connection with the semi-automatic type of turret lathes and automatic screw machinery, especially the latter. The screw, shown in the latter portion of its formation in position at Fig. 2 before it has parted company with the "parent" bar stock, is about to have the four holes drilled in the head; this screw, finished, is shown at Fig. 5. Referring to Fig. 2, the turret carrying the tool is supposed to have traveled up to the half-finished screw, which only requires threading and cutting-off to complete its manufacture as far as the "automatic" is concerned.

The shank of the screw enters the tool-steel spring collet *F* and butts against the adjustable stop *H*, forcing the collet inward and causing it to grip the screw tightly. The rotary motion of the bar is th is transmitted immediately to the collet and from thence to the machine steel bearing sleeve *B* by means of the tool-steel keys *K*. Play is allowed for the keys in *F* and *B* to compensate for both the butting and subsequent withdrawing movements of the turret. Keys *K* are similar to those denoted by K_2, shown at Fig. 4.

part of Fig. 2 which is let a short way into both the carrier head and the machine-steel driving bevel pinions *M;* the bevels rotate the carriers by means of key *P*, the slot in the carrier running out at the tail end. These bevel pinions are carried by the sleeve *B* and rotate bodily with it, planet-wise, receiving revolution about their own axes by gearing with the fixed bevel *J*, which is of phosphor bronze and acts in addi-

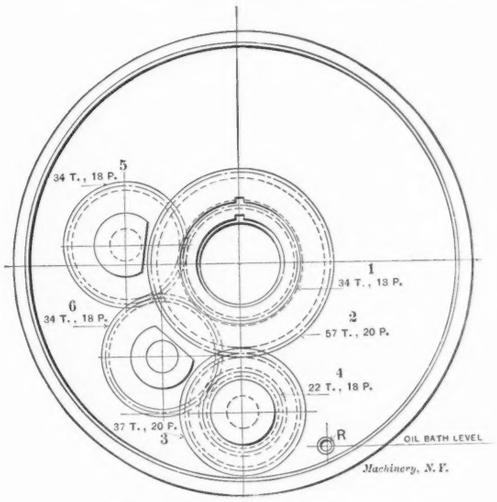
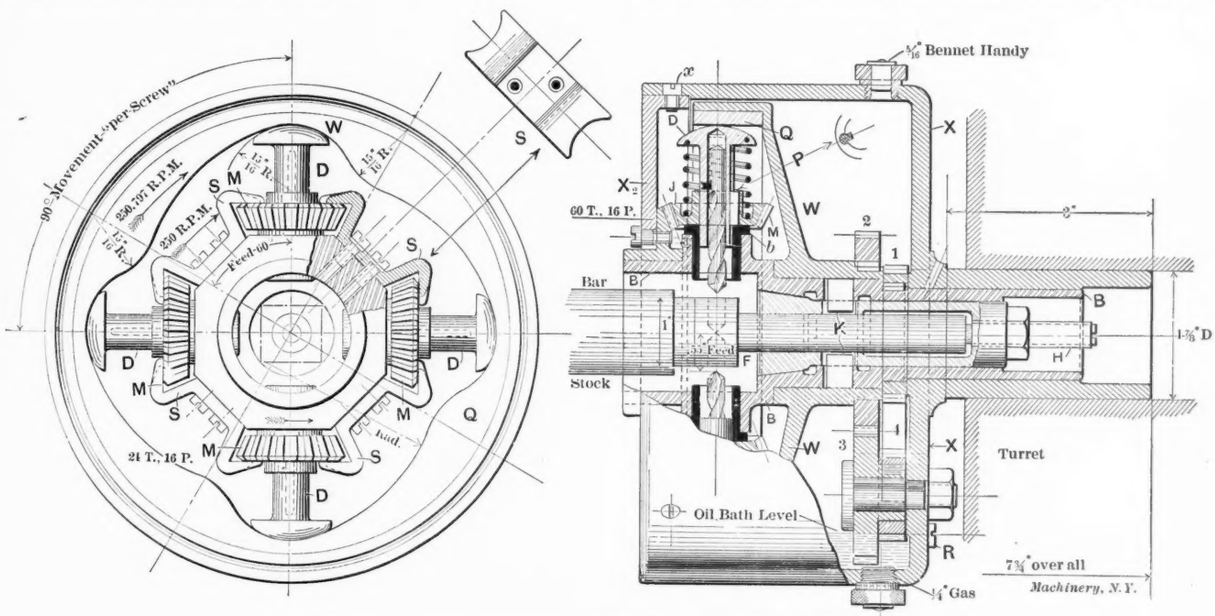


Fig. 3. Geared Drive of Feed Cam.

tion as a sort of steady bearing for *B*. *J* is secured to the gray iron end plate X_2 by means of four cheese-head screws. The bearing part is lubricated by a small hole drilled at an angle to the surface. It will now be seen that on the turret gaining position, the drills will revolve rapidly. The feed is the next and altogether the toughest problem to digest.

The feed is obtained by the cam *Q*, which is of hard tool steel, driven into the dished casting, *W*. One-sixth of a revo-



Figs. 1 and 2. Drilling Attachment arranged for Drilling Capstan Screw Heads.

Referring to the end view, Fig. 1, it will be seen that *B* carries an octagonal shoulder solid with itself. Holes are reamed through any two pairs of opposite flats, to admit of the four hardened cast-steel bushes, *b*, Fig. 2, which are a good fit and are secured by grub screws. These bushes act as bearings for the non-hardened high-grade tool-steel drill-carriers *D*. The carriers are reamed out to take the drills, which are high-speed steel and of the ordinary straight shank type with the exception of a slot cut to take the holding grub-screw, the slot allowing for adjustment of drills for grinding, wear, etc. Each carrier is held in position when the tool is out of action by the weak spring shown in section in the upper

lution of *Q* completes the feed and on the full one-fourth revolution being obtained, the drills have sprung back and everything is in readiness for another cycle. That is to say, the dished casting *W*, and consequently the cam, of course, must travel slightly faster than the sleeve *B;* this is effected by the train of gears shown in Fig. 2 and the end view, Fig. 3. No. 1 gear is keyed to *B* and drives through to 4 by the intermediates 5 and 6; 4 and 3 are fast together, and consequently rotate 2, which is keyed to *W*. One rotation of sleeve *B* gives *W* 629/627 turn; hence, with *B* running at 250 R. P. M., the gears turn the cam 0.797 revolution per minute faster than the sleeve and drills. The feed, as shown by Fig. 1, is completed

when cam *Q* has turned through 60 degrees in advance of the drills, sleeve, etc.; the remaining part of the full 90 degrees is taken up in allowing the drill carriers to slip back ready for the bar to withdraw (or rather for the turret to recede). The drills depend greatly on the centrifugal force to return them, the springs being weak. The rate of feed to the drills in this case runs to about 2.625 inches per minute, the actual distance fed being 0.55 inch. Taking the bar to run at 250 R. P. M., and with the fixed gear *J* having 60 teeth and the pinions 24 teeth, the drills will run at 625 R. P. M., which speed and the before-mentioned feed are averages for good high-speed steel drills (in mild steel only).

The peculiar shaped castings *S* are of phosphor bronze, and serve to keep the pinions from getting out of gear under the action of the centrifugal force exerted on them by reason of their planetary motion.

The drills are lubricated by an inclined jet of lard-oil in the usual manner. The cuttings from the drills will be found to take care of themselves, providing the relative sizes of bar and aperture are of a similar proportion to that in the sketch, being partly rolled, drawn, and partly shot out by the rotary movement of the turret.

This tool, with the addition of a few liners and adjusting screws, like *H*, will take any size of screw below the maximum

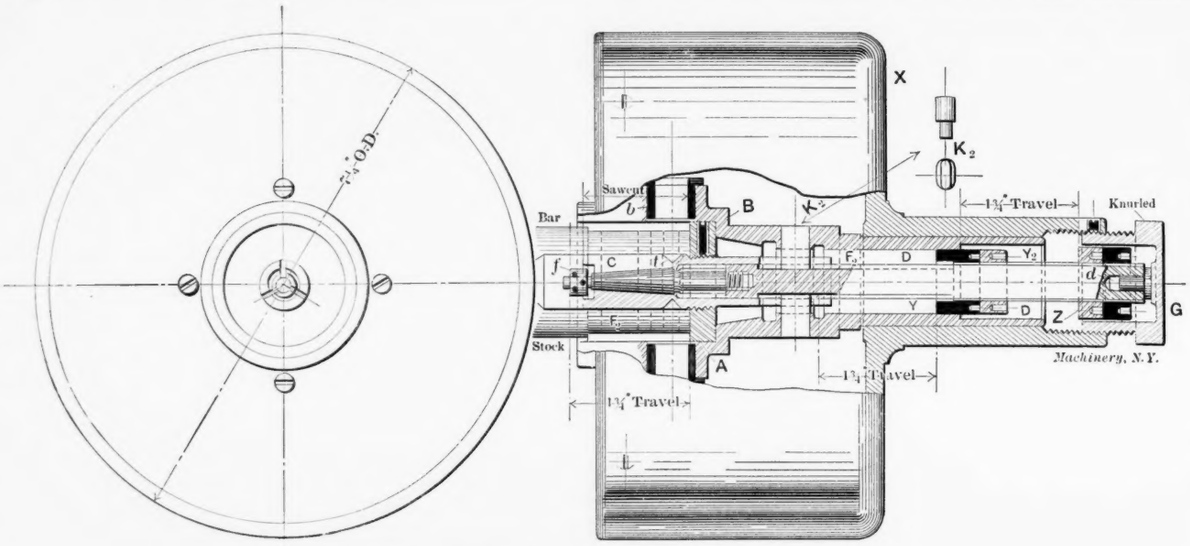


Fig. 4. Drilling Attachment arranged for Drilling Capstan Nuts.

This cam motion, while being delightfully simple and certain, has the objection (if it be one) that the holes are not drilled completely through the head. To do this, a cam should, of course, be designed which would give two of the drills intermittent feeds after the other two had gone to a similar depth to that of the present tool; this would remove the small amount of metal left. This proposed cam, however, would necessarily be complicated and would require a "reverse" setting after the manner of a certain class of self-opening die-head, thus doing away with the absolute automatic action of the tool as it is at present. But I cannot see that it is a very important failing, as one seldom puts a tommy-bar further in the hole than the depth shown as drilled in Fig. 5.

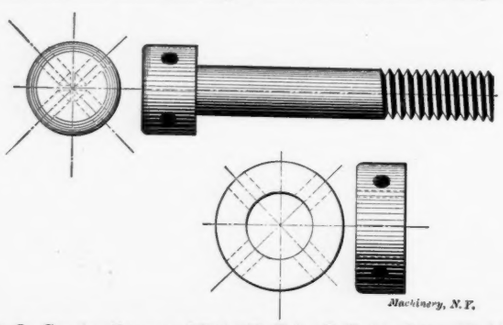


Fig. 5. Capstan Screw and Nut with Holes Drilled in Screw Machine.

The maximum diameter of screw-head this particular tool will take for a given outside over-all diameter of 7¼ inches, is a little more than one inch. The shank of the gray iron body could be reduced somewhat if cast in some tougher metal, such as "Harrison's steel alloy," a product of Lincoln, England, which combines the usual qualities of a steel casting with excellent machining and bearing features. Lubrication of the gears and cams is effected by an oil bath which can be replenished or drawn off by the two machine-steel plugs shown at Fig. 2, which are further fitted with "Bennet Handy" oilers. The bath is tested for quantity by the examining plug *R*, the center of which is the oil-bath level. See Fig. 2. Fig. 1 shows the bevels and carriers with springs and drills removed.

given. The construction, though seemingly complicated, is fairly simple; it is very easy to set, for to withdraw the drills, sleeve, etc., one has only to remove the four screws *x*, when the whole works can be drawn out, leaving only that part of the gear train which is secured to *X*. The drills can then be removed at once by slacking the grub screw in the carriers. It will be apparent that the number of drills may be three, four, five, six, or even eight, should requirements demand it; usually, however, four are adequate.

Fig. 4 shows the same tool fitted up to drill the nut shown at Fig. 5. In this case the hole (which should have a rose-reamer run through in a previous operation) is not yet tapped so that it may be gripped by the internal spring collet F_2. On this is screwed the adjusting butt collar *A*, of which one is required for each thickness of nut. The collar is tapped and locked on the screwed part of F_2 by a brass grub-screw. The groove *t* on F_2 allows the drills to make clear holes. The hardened cast-steel plug *C* screws into the arbor *D*, which fits without shake into the end of F_2. Key slots are cut in both *D* and F_2 to receive the driving keys K_2. The further end of *D* has a fine thread, on which is the bearing nut *Y* and its lock-nut Y_2, which adjust to suit the varying widths of nuts; these last only serve to steady the arbor *D*, which is prevented from being withdrawn by nut *Z*. Driven into the end of *D* is the tool-steel plug *d*, which butts against the knurled shell-nut *G*, which is adjustable to suit the exact travel of F_2 on *C* after the conical surfaces come in contact. The work should butt up on *A*, and the pin *d* on *G*, simultaneously.

This adaption of the tool enables it to clear the cuttings out by the withdrawal of the turret, and, as shown, the "clearing" is accomplished in 1¾-inch travel. It will be observed that with the exception of the bushings *b*, the sleeve *B* is pecisely as in Fig. 2.

* * *

Mr. C. W. Shelley, Wallingford, Conn., advises us that cornstarch is preferable to chalk in the receipt for die-makers' wax published in the October issue of MACHINERY. The use of chalk was recommended to prevent stickiness, but cornstarch works better as it makes the grain of the wax finer, thus showing up lines more distinctly.

THE "LO-SWING" LATHE.

A glance at the accompanying cuts will be sufficient to convey the information that this lathe, called by its makers the "Lo-Swing," is very far from being a representative of standard practice. Its builders, the Fitchburg Machine Works, Fitchburg, Mass., set out with the idea of making it a single purpose machine. Provision is made for turning work from 3½ inches in diameter down, of any length up to 60 inches, and they have made provision for nothing further. It will not take work of larger size, it cannot be used as a chucking machine, nor can it cut a thread without special attachments not regularly furnished. A little thought will convince any one that by far the greater part of the lathe work in the ordinary machine shop falls fairly within these limitations, and if a machine is designed to work within these dimensions and take care of every possibility for improvement which this range allows, it should be able to do its own work better than can the standard machine.

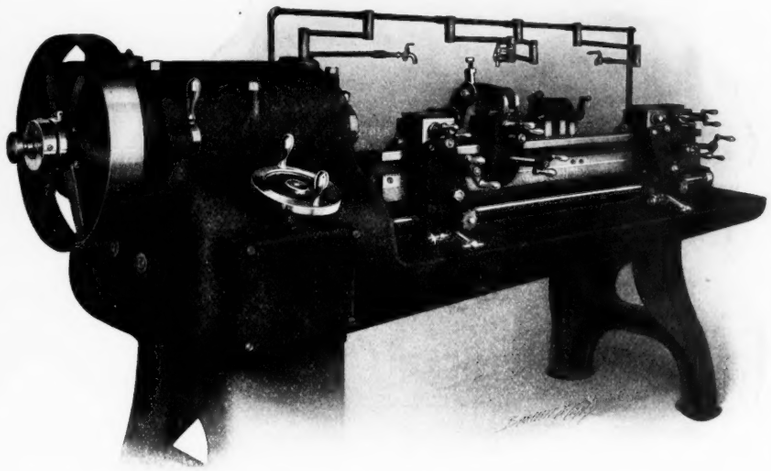


Fig. 1. Front View of Lathe at Headstock End.

Figs. 1 and 2 show the general arrangement of the lathe. The bed, oil pan, and headstock are cast in one piece. The machine is driven by a fast-running belt on a 12-inch pulley, from which it is never shifted, the seven spindle speeds furnished being obtained through gearing in the headstock. Fig. 4 is a top view of the headstock with the cover removed. In the hub of driving pulley *A* is mounted a bolt, *B*, which may be shifted by its handle to engage latch pins in either collar *C* or gear *D*. When in engagement with collar *C*, the spindle, through it, is driven directly; when in engagement with gear *D*, which is loose on the spindle, motion is transmitted through gear *E* to auxiliary shaft *F*. A pinion integral with this shaft, and two gears *L* and *O* mounted on it, drive respectively the three gears, *G*, *K*, and *N*, which thus revolve loosely on the spindle at varying rates of speed. If knob *H* at the left-hand end of the spindle be pushed inward, collar *J*, which is keyed to the spindle, will be moved to the right, engaging the driving pin *I* with the latch pin in gear *K*, thus locking this gear to the spindle. In a similar manner, by pulling out knob *H*, gear *N* will be locked to the spindle. If collar *J* is in a central position, as shown in the cut, both *K* and *N* will be disconnected and the spindle will be driven by gear *G*. This gear has a ratchet arrangement, not clearly shown in the cut, by which it normally engages flange *Z*, which is pinned to the spindle. If, however, the spindle is connected at any time to a higher speed gear, this ratchet, which is noiseless, will allow the spindle to take a faster gait. A handle which may be seen at the front of the headstock in Figs. 1 and 2 throws in back gears *Q* to transmit the motion from pulley *A* to gear *D*. All these gears run in oil. The seven speeds given by this mechanism range between 50 and 560 R. P. M. The changes cannot be made while the machine is running, but it was deemed advisable to sacrifice this point to secure simplicity of design. The live center is in the form of a button, set in a recess in the end of the spindle. The tail of the dog is driven by a stout pin, which passes through the spindle nose.

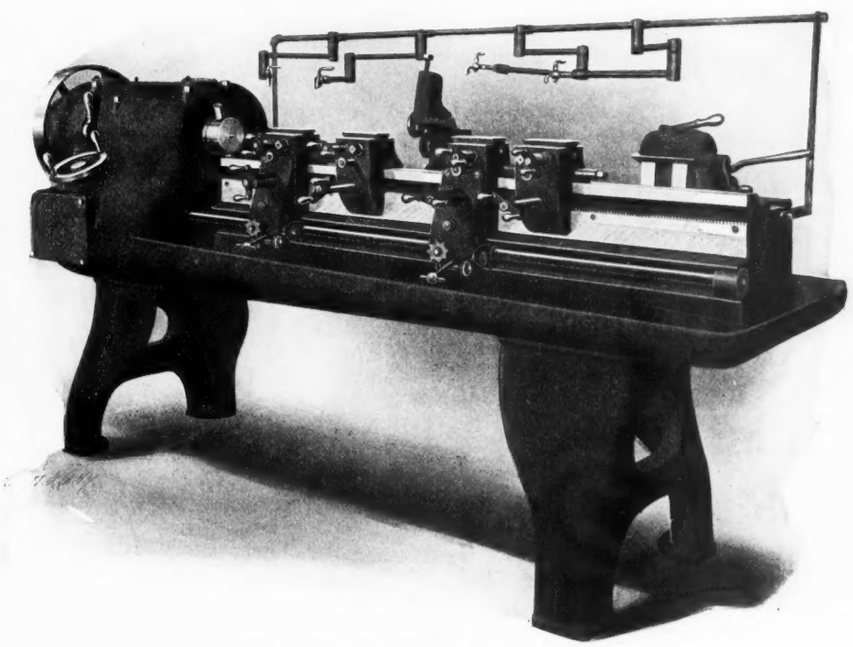


Fig. 2. Front View of Lathe from Tailstock End.

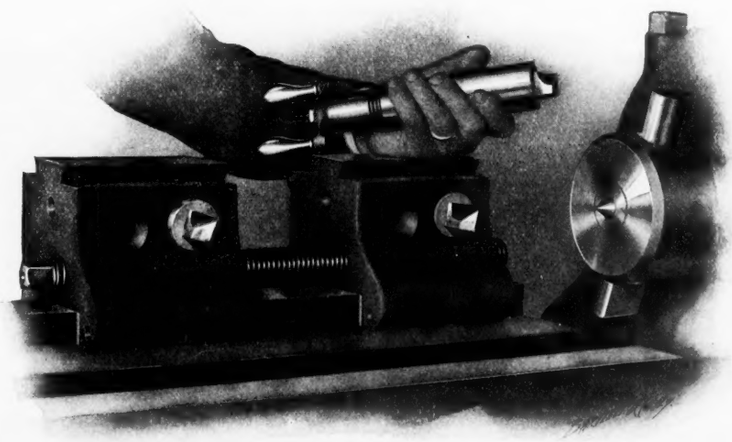


Fig. 3. The Carriages and Toolholders.

The chief interest of the machine centers

around the arrangement of the carriages, and the tool and work supports. Fig. 3 gives a good idea of the design of the carriage and the rigidity which is obtained by bringing the ways on which it slides up as near as possible to the point of the tool. There is a great difference between the degree of rigidity which is attained by this construction and that possible in an 18 or 20-inch lathe of usual design, especially if the latter is furnished with a compound rest, in which case the point of the tool may be anywhere from 8 to 15 inches from the guiding surface of the V's, with the connection between them made up of a multiplicity of joints and sliding surfaces. Here the cutting point is directly over the bearing and the number of fits between the tool and bearing is reduced to the smallest possible number. The tool itself is machined to fit in a sleeve, which may be seen in the hand of the workman in Fig. 3. The inner end of the cross motion screw bears on the end of the tool, giving an absolutely direct action. A yoke engages notches at the end of the tool and thus draws it out when the screw is turned back. The sleeve with this leadscrew and tool is put in through one of the two holes shown in each carriage. In the other hole is placed an eccentric clamp which binds the sleeve in any position desired. As may be seen from Fig. 2 there are four carriages furnished regularly with the machine of which two only have power feed. Each of the carriages with feed carries another with it at a distance which may be regulated by turning the adjusting screw which connects them. On a shaft long enough to allow it, four different cuts may thus be taken at once; two in one direction and two in the other, if conditions warrant such a proceeding. Since they are only about 5 inches wide, the tools in the adjacent carriages may be brought very close to each other, and their light weight allows them to be handled with extreme ease. Variation in feed is obtained by turning the double handled wheel which may be seen at the front of the headstock. This operates a modified form of Sellers' friction drive.

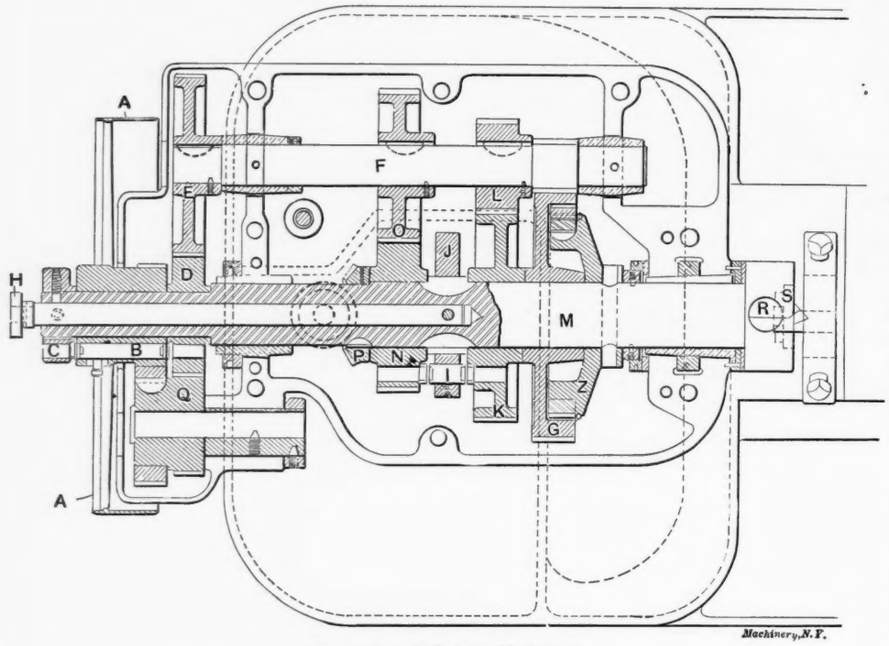


Fig. 4. Spindle Driving Mechanism.

Fig. 5 is a line engraving of the tailstock. It is mounted, as shown, in separate ways on the rear of the bed, and is of such form that it offers no interference to the movement of the carriages,

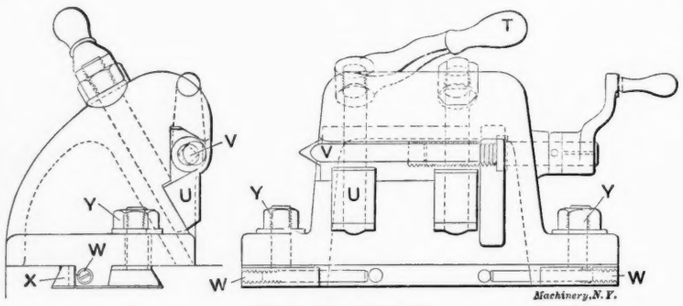


Fig. 5. The Tailstock.

which may be moved back and forth across its face at will. Handle *T*, through hook bolt *U*, serves to clamp the triangular body of the center *V*. A crank is furnished for moving the center in and out. Adjusting screws, *W*, give a longitudinal movement to the taper gib, *X*, thus serving to line up the center laterally with the spindle. Nuts *Y* clamp the tailstock in place by obvious means.

The back rest and follow rest idea, which has been found so useful in screw machine practice, has been adapted to suit the design of this machine. Fig. 1 shows a follow rest clamped to the ways provided for it on top of the tool carriage. A similar center rest is shown in Fig. 6, which is a cross section of the bed near one of the main carriages. This follow rest is clamped to the surface on which the tailstock slides, and like the tailstock, offers no interference to the carriage. Owing to the high speeds at which the work may be run with modern tool steels, the usual form of stationary jaws for the follow and center rests has been discarded, and a roller rest introduced in its place. A neat arrangement will be noted in the design of the adjusting screw for these follow and center rest jaws. It is made in two diameters, one of which is threaded right-hand and the other left-hand, screwing respectively into the main casting and jaws. This construction does away with the necessity for a collar.

One result of the rigid support which the design of this lathe gives to the tool, is the fact that the cutting edge lasts much longer and may be given a much sharper angle than is possible with a standard lathe. It is well understood nowadays that the great destructive factor at work on the cutting edge, is not the wear due to the friction of the

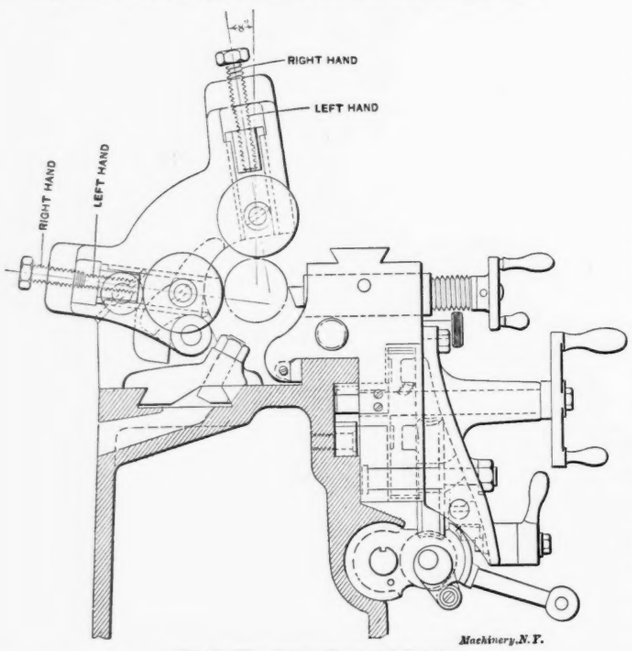


Fig. 6. The Center Rest and Carriage.

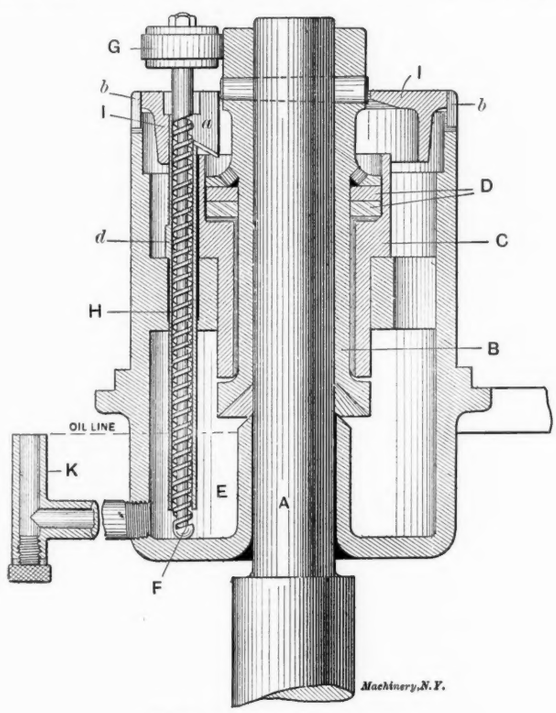
chip. When a heavy cut is being taken the chip is split off from the stock a perceptible distance ahead of the tool and the edge itself often does not come in contact with the chip, which strikes the tool considerably to the rear. This action is especially noticeable in cutting steel, when a perceptible groove is ofttimes worn just back of the cutting edge. The factor that breaks the edge of the tool, is the lateral vibration which is allowed by the usual tool post and carriage. It is claimed to be possible, with a lathe constructed in the manner of this one, to carry a much sharper cutting angle than heretofore, since the rigidity of the design almost eliminates the lateral vibration and thus makes it unnecessary to blunt the angle of the tool edge to keep it from breaking. A sharp cutting edge will of course require much less power and consequently work with less friction and heat than will a blunt one.

This machine was designed by Mr. James Hartness, of Springfield, Vt., as may have already been surmised by those familiar with the general lines and details of the flat turret lathe. Mr. Hartness, however, has no official connection with the Fitchburg Machine Tool Co.

* * *

AUTOMATIC OILING DEVICE FOR ARMATURE BEARINGS ON VERTICAL MOTORS.

The problem of oiling armature shaft bearings on a vertical motor is not only to keep the bearings well lubricated, but at the same time to prevent oil from getting on to the commutator and armature windings. In a vertical motor the bearings must take care not only of the side pressure of the shaft, but the end thrust due to the weight of the armature shaft and attached parts as well. The Northern Electrical Mfg. Co., Madison, Wis., use an ingenious and simple combination in connection with their line of vertical motors which is said to effectually fulfill the requirements outlined in the foregoing.



Oiling Device for Vertical Bearing.

In the sectional view of the bearings shown in the accompanying cut, *A* is a portion of the armature shaft reduced for the journal; the journal proper, however, is the bushing *B*, which is formed at its lower end so that the excess oil runs off into the reservoir without tending to follow the shaft. The bushing is pinned to the shaft at the upper end, and is fitted to the brass or bronze bearing *C*, which latter is formed into a seat at the upper end to contain the thrust washers *D*. A corresponding flange is turned on *B* for the thrust bearing, and is perforated with a number of oil holes for admitting the lubricant. The upper end of *B* is turned to form a driver for the friction roller *G*, which is attached to the upper end of a screw conveyer. This conveyer runs inside of a tube, which is loosely fitted to the main casting *H;* to the upper end of the tube is snugly fitted the cover casting *I*. The oil brought up through the tube is discharged by spout *a* into the cup of the thrust bearing where it passes through the oil holes into the thrust bearing and from thence into the journal and reservoir *E*. In this manner a continuous supply of oil is furnished the bearing, while the motor is running; in fact, it is flooded with oil so that the best possible lubrication is always obtained.

In order that the friction roller *G* may be held against the driving collar with sufficient pressure to drive it at all times, a simple and ingenious expedient is applied. The cover casting *I* is eccentrically supported so that the side thrust pushes the roller *G* against the driving collar. The cover casting is kept in position sideways by two projections *b*, which fit into notches in the upper edge of the main casting. It is carried by the conveyer tube which loosely fits in the casting at *H* and rests upon a lug on one side at *d*, thus supporting it on one side only so that its weight rocks it over until the roller *G* comes into contact with the driver. These parts form a unit and may be lifted out at any time for inspection, without removing screws or other fastenings.

When the motor is to operate in both directions, the conveyer is cut with both right- and left-hand screws, and even though the continuity is broken and it would seem that one screw would tend to undo the work of the other, sufficient oil is carried up to insure thorough lubrication at all times. In order to prevent too much oil being placed in the reservoir, an open filling tube *K* is used, which serves at the same time as a means of gaging the depth of the oil in the reservoir.

* * *

THE CREATION OF DRAFT.

A treatise on mechanical draft published by the B. F. Sturtevant Co. says that although the ultimate object of any means of draft production must necessarily be to create draft or velocity sufficient to provide the required amount of air and to carry off the gases, yet this portion of its work is almost infinitesimal as compared with the demand made for sufficient pressure to overcome the resistance of the fuel and the boiler. In other words, the ability to create sufficient pressure difference is the primary requisite for burning a given quantity of fuel, rather than the ability to move a certain amount of air. Draft-producing apparatus is not, therefore, to be based merely upon the total number of cubic feet to be moved per hour, as determined by multiplying the coal consumption by the allowance of air per pound of coal. If this were the case, a low chimney, or a large slow-running fan, would meet the requirements. In reality, the relatively immense resistances of fuel and boiler demand that the chimney or fan shall first be designed to create sufficient intensity of draft to overcome these resistances and to create the requisite velocity. This velocity must be such that, if multiplied by the full area at which it is measured, the product will equal the volume of air necessary for the combustion of the stated amount of fuel. The height of chimney or the diameter and speed of fan necessary to create the draft thus shown to be required having been determined, it is only necessary to make the capacity such as to accommodate the given volume of air.

* * *

Since the advent of high-speed steel, one of the stock jokes concerns the man who used a high-speed drill for ratchet drilling and then complained that it did not cut any faster than the ordinary kind. But, nevertheless, it appears from experiments that have been made that high-speed drills are superior for drilling rails with a ratchet or hand-drilling machine, to the carbon steel drills. The gain is not, of course, because the high-speed drill cuts faster, but because it can be used so much longer without regrinding. The Rich flat drill, a special form of high-speed drill for rail drilling, has drilled 1,600 holes one inch in diameter in 80-pound rails without being reground, whereas 50 holes with the ordinary twist drill would have been considered a good performance. Another interesting comparison has been made in the New York Navy Yard, where one of these drills drilled ten 1¼-inch holes in turret armor plate without re-sharpening, while eight twist drills of carbon steel were used for finishing one hole.

LETTERS UPON PRACTICAL SUBJECTS.

HEROIC METHODS IN THE MACHINE SHOP.

Editor MACHINERY:

Reading the account of the repairs to a hydrostatic press in the August MACHINERY, I was reminded of a remark I once heard made by an old man who was foreman of a blacksmith shop: "Only two blacksmiths ever went to hell, one for working with hot tongs, and the other for hammering cold steel."

We see the results of such "heroic" methods in every machine shop. Holes are drilled in the drill tables, the ways of the lathes are cut and the *heroes* are using the planer platens for an anvil with a monkey wrench for a hammer. They probably "get there," but they used a Stilson wrench on finished brass pipe and set up the nuts with a hammer and chisel. While there are some men who are unnecessarily cautious, a larger number will be found to be more or less "loose" in their methods.

In the repair job mentioned, if the man who turned the fits had taken a little more time in measuring his work perhaps he could have gotten his work together without breaking his press. If Mr. Davis had squared the end of the bronze valve stem in a lathe perfectly straight and smooth, and had made a good seat, the valve would have been tight enough for all practical purposes. If the valve leaked it would not take half an hour to grind it. The "sledge hammer" method was a dangerous proceeding; there was danger of bending the bronze rod in the middle, causing the end to bear unevenly on the seat and leak anyway.

At one time I was fireman in a power station. The gage cocks were of the well-known type, opened by pulling a chain and closed by a weight on the end of the stem.

One day one of them leaked quite badly, and my "opposite" in coming on duty remarked that he would fix that "right off quick." He gave the chain a tremendous yank; it parted in the middle. I left him pounding on the weight with a hammer. It is unnecessary to state I found the gage cocks still leaking when I came on duty again and they leaked till I had a chance to reduce the steam pressure to two or three pounds and take the valve out and face off the end.

I think the old proverb: "What is worth doing at all is worth doing well," applies to machine work as well as other things. If any of the readers doubt this, they should hunt up their old reading books they had when they went to school and read the old story they used to read about "Jim! Jim! The pig is out!"

Portland, Me. H. K. GRIGGS.

PROPORTIONS OF COUNTERBORES.

Editor MACHINERY:

I saw an article in the August issue of MACHINERY pertaining to twist drills. I thought a great deal of the way in which the formulas and table were put up, as well as of the general information given in regard to the making of such drills. As tables of like character must be of great value to toolmakers and manufacturers, I herewith submit a similar table with formulas and a few remarks in regard to the making of counterbores

Dimensions have been given in relation to the diameter of the body of the counterbore *A*. As counterbores are used as well with straight shanks as with taper shanks, I thought the simplest and most universal way of putting up the formulas and table would be to do so with reference to straight-shank counterbores only; but as there is no reason to make the total lengths different for counterbores with straight or taper shanks, the formulas will also hold good for counterbores with taper shanks of reasonable proportions, as will also the dimensions for *B* and *C*. As the Morse standard taper shanks are more used than any other standard taper shanks, for the convenience of the readers I herewith give an auxiliary table giving the numbers of Morse taper shanks that ought to be used with certain size counterbores.

Diam. of Body, inches.	No. of Morse Taper.
¼—½	1
17-32—⅞	2
29-32—1⅜	3
1 13-32—2	4
2 1-32—3	5

In the following formulas:

A = diameter of body.
B = length of body.
C = length of guide.
D = length of shank.
E = diameter of shank.
F = diameter of neck.
G = total length.

For counterbores from ¼ to 1 7-16-inch the following formulas should be used:

$$B = \frac{4A}{3}$$

$$C = \frac{3A}{4}$$

$$D = A + 2\tfrac{1}{4}.$$

$$E = A.$$

$$F = A - 1\text{-}32.$$

$$G = 7A + 3\tfrac{1}{8}.$$

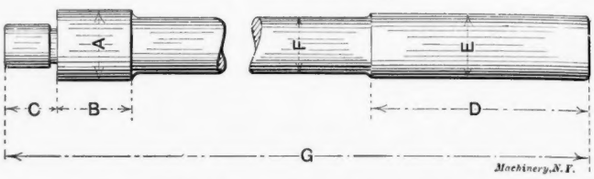


TABLE OF COUNTERBORE DIMENSIONS.

A	B	C	D	E	F	G
¼	11/32	3/16	2 1/32	¼	7/32	4 7/8
5/16	13/32	1/4	2 9/32	5/16	9/32	5 5/16
3/8	15/32	9/32	2 5/8	3/8	11/32	5 3/4
7/16	17/32	5/16	2 11/16	7/16	13/32	6 1/16
½	19/32	3/8	2 3/4	½	15/32	6 5/8
9/16	21/32	27/64	2 13/16	9/16	17/32	7 1/16
5/8	23/32	15/32	2 7/8	5/8	19/32	7 1/2
11/16	25/32	33/64	2 15/16	11/16	21/32	7 15/16
¾	1	9/16	3	¾	23/32	8 3/8
13/16	1 3/32	39/64	3 1/16	13/16	25/32	8 13/16
7/8	1 5/32	21/32	3 1/8	7/8	27/32	9 1/4
15/16	1 7/32	45/64	3 3/16	15/16	29/32	9 11/16
1	1 1/4	3/4	3 1/4	1	31/32	10 1/8
1 1/16	1 13/32	51/64	3 5/16	1 1/16	1 1/32	10 9/16
1 1/8	1 7/16	27/32	3 3/8	1 1/8	1 3/32	11
1 3/16	1 19/32	57/64	3 7/16	1 3/16	1 5/32	11 7/16
1 1/4	1 21/32	29/32	3 1/2	1 1/4	1 7/32	11 7/8
1 5/16	1 23/32	61/64	3 9/16	1 5/16	1 9/32	12 5/16
1 3/8	1 27/32	1 1/32	3 5/8	1 3/8	1 11/32	12 3/4
1 7/16	1 29/32	1 5/64	3 11/16	1 7/16	1 13/32	13 1/16
1 ½	2	1 1/8	3 3/4	1 ½	1 ½	13 3/8
1 5/8	2 3/32	1 7/32	3 15/16	1 5/8	1 17/32	13 3/4
1 ¾	2 3/16	1 5/16	4	1 ¾	1 19/32	14 1/8
1 7/8	2 9/32	1 13/32	4 5/16	1 7/8	1 21/32	14 1/2
2	2 ¼	1 ½	4 ½	2	1 25/32	14 7/8
2 1/8	2 15/32	1 19/32	4 11/16	2 1/8	1 29/32	15 1/4
2 ¼	2 9/16	1 11/16	4 7/8	2 ¼	1 ¾	15 5/8
2 3/8	2 21/32	1 25/32	5 1/16	2 3/8	1 25/32	16
2 ½	2 ¾	1 7/8	5 ¼	2 ½	1 27/32	16 3/8
2 5/8	2 27/32	1 31/32	5 7/16	2 5/8	1 31/32	16 3/4
2 ¾	2 15/16	2	5 5/8	2 ¾	2	17 1/8
2 7/8	3 1/32	2 5/32	5 13/16	2 7/8	2 1/32	17 ½
3	3 1/8	2 ¼	6	3	2 1/16	17 7/8

For counterbores from 1½ to 3 inches the following formulas should be used:

$$B = \frac{3A}{4} + \tfrac{7}{8}.$$

$$C = \frac{3A}{4}$$

$$D = \frac{3A}{2} + 1\tfrac{1}{2}.$$

$$E = \frac{A}{3} + 1.$$

$$F = \frac{A}{3} + 15\text{-}16$$

$$G = 3A + 8\tfrac{7}{8}.$$

As will be seen, no dimension has been given in the table for the diameter of the guide, the reason for which is obvious; the dimension for this must, of course, be determined by the size of the hole drilled before counterboring.

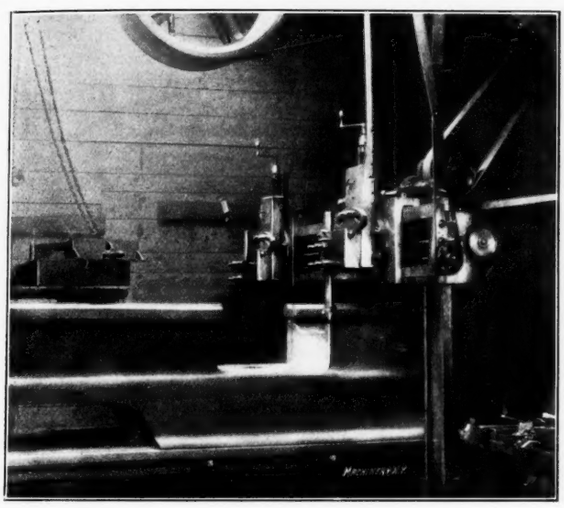
Counterbores are generally made with four flutes. As a rule the flutes are cut spiral, but if counterbores are to be used on brass or cast-iron better results will be obtained by fluting them straight. When milling the flutes in spiral fluted counterbores, the helix should be so selected as to make the lead $= 12 \times$ diameter. The angle of the flute with the center line will then amount to about 15 degrees.

New York. A. ANDERSON.

CROSS PLANING A GAP-LATHE BED.

Editor MACHINERY:

We recently had occasion to machine a heavy 38-inch lathe-bed casting 27 feet long, weighing 10,000 pounds, which had a removable section 26 inches long, making a gap in the bed for turning large diameters. The fitting of this gap piece necessitated planing crosswise of the bed. We had been in the habit of planing beds of this description up to 18 feet in length by bolting the casting crosswise of the planer, allowing the long projecting end to travel on a greased parallel support. The extreme length and weight of this casting necessitated some other expedient. Referring to the accompanying photograph, directly over the crossrail of the planer was placed a friction countershaft having two speeds, a forward and a quick return. This counter shaft was operated by hand by means of a shipper conveniently located. A small pulley was substituted in place of the feed pinion on the cross-feed screw, to take the power from the countershaft. The tool used was the ordinary undercut planer tool. W. L. FAY.



A Job of Cross Planing.

Dexter, Me.

AUTOMATIC SLIDE FORMING PUNCH.

Editor MACHINERY:

The automatic slide forming punch and die shown in the illustration is used to form the tin blank to the shape shown in Fig. 3. As this punch and die is of a rather intricate and elaborate type, and the work produced is of superior quality, a detailed description of its principal working parts may be of interest. The material used is soft tin 0.012 inch thick, the blank being first cut to the required dimensions in the shears by the use of suitable gages and stops. We used this die in a press of the inclinable pattern, thus affording the operator an opportunity to continually feed the blank to the die; the finished work spreads slightly as the punch ascends, and slides off by gravity into a receptacle at the rear.

It may be interesting to note, on referring to the sketches, the means provided for operating the sections which form the work on the sides. It is plainly evident that the slides which make the side bends can be made to move in more or less, to a limited extent, at the will of the operator, by adjusting the studs on the punch that engage the bell-crank levers on the die; these studs, held by check-nuts, are adjustable, having ample range for this purpose.

While the drawing shows only the front view, the same method of construction is employed at the rear, there being in the make-up of the die four studs, *L*, and a like number of bell-crank levers, *D*. Figs. 1 and 2 are front views of the die and punch respectively; Fig. 3 a section and plan of the finished work. In the die, Fig. 1, *A* is the cast-iron base, *B* the tool-steel slides secured on the face of *A* by four shoulder screws, *C*, two of which are shown. Rectangular slots that fit the body of the screws sidewise are provided in the slides to permit them to slide back and forth more than the necessary distance.

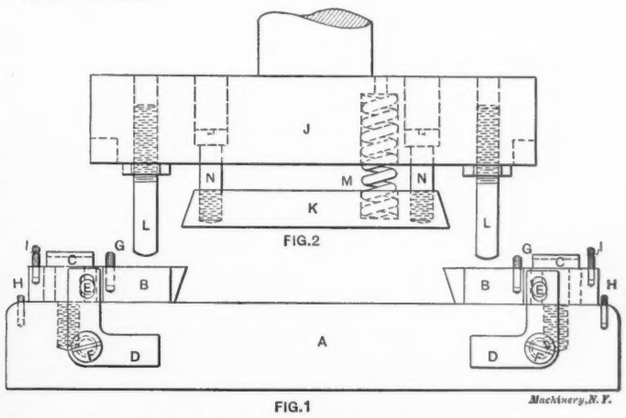


Fig. 1 and 2. A Slide Forming Punch and Die.

The levers, *D*, are pivoted at *F*, and through them the pins, *E*, project, their inner ends having been driven into *B*. These act on the latter when the punch descends and forces the slides in. The pins, *G*, gage the blanks sideways, and *H* check the outward action of the slides. The die, after assembling, was screwed and doweled to a plate of larger dimensions, having pins standing up at the extreme sides, to which springs were hooked extending from the pins, *I*. These were to pull the slides out to their limit on the upward travel of the press ram.

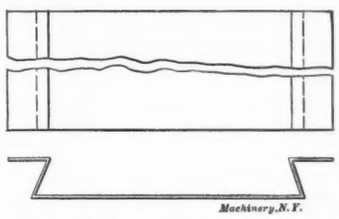


Fig. 3. Product of Die shown in Fig. 1.

The punch, Fig. 2, comprises the holder, *J*, of castiron, form *K*, of tool steel, and the studs, *L*. Part *K* is kept down by eight springs, *M*, arranged conveniently for this purpose, the number being varied according to the stiffness of the blank stock. The former is necessarily fully extended until the punch forces the blank to the bottom of the die, and four screws, *N*, limit its downward projection, one at each corner.

It is not necessary to describe the methods employed in making the various parts of the tool, the machine work being quite clear to any ordinary die-maker. The parts that need the greatest care in making are the former and slides. All sections, with the exception of the body and screws, are hardened, thus increasing the durability of the die. The tool is "subpressed" by means of two 1-inch pins which are fixed in the lower die and are a good sliding fit in the punch holder; for the sake of clearness these pins are omitted from the illustration. C. H. ROWE.

Pittsfield, Mass.

HIGH-SPEED DRILL FOR SPINDLE FORGINGS

Editor MACHINERY:

In drilling spindle forgings, we are often troubled with flaws and seams, some of them being full of sand. To run a twist or water-tube drill through such a spindle is not very satisfactory. In our experience, we would run into such a flaw about ¼ inch, when the drill would stick and break, or turn around with the work, and break off the hose connection. This of course was a little cheaper than breaking the

drill, so we made the connection loose enough for it to slip before the drill would break, but this did not always suffice, and after we had broken a couple of drills on one bad lot of spindle forgings, I went to work and fixed up a drill that the sand would not hurt.

I took a piece of 5-16 inch square "Rex" high-speed steel, the kind that is usually used in tool-holders, and cut it off about 3 inches longer than the piece to be drilled; one end was flattened to the required size, which in this case was 9-16 inch. The cutting lips of the drill were hardened by heating to a cherry red and dipping in coal oil. Care should be taken not to heat the piece too far back, as the oil is likely to get too hot if there is too much steel to be hardened. After grinding the drill to size and notching the cutting edges so as to break up the chips, I was ready to start. This drill did not stick in the flaws or sand cracks, and it cut as fast as any twist or tube drill that we ever used on this kind of steel. Since this experience we have made a good many tools of this kind, and have used them to drill nearly all our tool steel. One of them 11-16 inch diameter, we run 25 inches deep into forgings without any trouble whatever from the chips, or sticking. The spindles we drill are of the regular 50 point carbon steel forgings.

C. B.

PLUG AND RING GAGE—DRY VS. LUBRICATED.

Editor MACHINERY:

Does oil take up space? Can a one-inch plug enter a one-inch hole? These two questions the writer has heard argued till every man in the room had his "spell," the result being that at the end of the argument we were no nearer a solution than at the start. A reader of MACHINERY has inquired why it is that a one-inch plug will enter a one-inch ring when properly lubricated, but will not enter when dry. The writer has been asked to offer a solution, but must admit that the question can be better handled by those who are not in the ranks with us "greasy mechanics." The writer has in his possession a plug and ring gage that was made purposely to demonstrate this fact, *i. e.*, when both gages are dry they will not go together without driving, it being impossible to make the plug enter by hand. With ordinary oils as lubricants the plug will start into ring, squeak and lock solid, requiring driving to remove. But with a good heavy lubricant (so heavy in fact that it will float small metal chips), the plug can be easily pushed in and out of the ring by hand. Now, if the surfaces are true and the fit is such that it requires driving, how can oil cover the surfaces and allow the plug to enter at the same time?

Here is one suggestion that may or may not be along the line of solution. First, let us go back a little, for I want to carry the reader along carefully as to the manner in which these gages were made. They were turned nearly to size and hardened; then ground sufficiently to remove the scale. Each gage was then dipped alternately into a kettle of boiling water (until it acquired the temperature of the water, which caused it to expand) then was suddenly immersed into ice water, causing it to contract. This alternate heating and cooling, many times repeated, caused the molecules to twist and change position as various strains were removed. This operation of dipping in hot and cold water was repeated perhaps one hundred times, the result being that the gages were thoroughly settled or "seasoned."

The plug gage was then smoothly ground to within 0.0015 inch of finished size. Then an adjustable cast iron lap and very fine emery paste was used to bring the gage to within "two tenths" of finished size. This lap was then superseded by a new one, perfectly free from emery, and the gage thoroughly cleaned. The emery used for the last "two tenths" was obtained by allowing flour emery to settle in a glass of oil and skimming off the top, using only the emery that floats, which must necessarily be very fine. Bringing a piece of steel to size with emery so fine that it floats will appear to some machinists in the same light that the Indian did to the traveler. The Indian was rubbing away at an old plowshare with a whetstone. The traveler asked him what he was doing. "Ugh! make arrow head." The gage was finished by lapping lengthwise by hand, the lap fitting snugly at all times. When finished, the gage had that peculiar color closely resembling burnished silver.

This pair of gages, remember, was made expressly to demonstrate the difference when lubricated and when dry, and naturally the greatest care was exercised in obtaining a true surface, and under an ordinary glass it was apparently true. But before we lubricate the plug to enter the ring, let us examine it under a Baush & Lomb microscope, magnifying 87 times. The plug gage now appears like a fine rat-tail file. It is my belief that these minute ridges act as tiny reservoirs and carry oil in to the center of the ring where it certainly lubricates, because the plug can then be easily entered in the ring. The fact that expensive chronometer oil, vaseline, lard oils, etc., will not lubricate sufficiently to allow the plug to enter, while a good machine lubricating oil will, should prove a lesson to those who squirt a drop of oil on the end of a finger, smell of it, mutter "lard," and continue to oil up with it. The readers should take up this question and argue it out through the columns of MACHINERY, that we may derive benefits from various minds.

PERPLEXED.

[The question suggested by our correspondent is one that has caused a great deal of perplexity and argument among tool makers and others acquainted with the peculiarities of a "gage fit." It is paradoxical that a plug will enter a ring gage easily when thickly oiled, but will not enter when dry. This condition would apparently indicate that the film of oil occupies no space, or that its presence makes the ring larger, and the plug smaller. It has been suggested that perhaps the oil has a smoothing effect on the surface particles of the gage and plug, similar to that, for example, of the hand on the fur of a cat. Another suggestion is that surfaces brought to the high perfection of finish described by our correspondent, tend to cohere when brought into intimate contact. It has been noted that accurately fitted end gages say ¼ inch square, require an appreciable force to separate them, more in fact, than can be attributed to atmospheric pressure alone. This would seem to indicate that initial cohesion or some other attractive force exists between accurately fitted surfaces, but cohesion means that the surface particles interlock, and this they cannot do in the case of a plug and gage without an extension beyond the normal dimensions. The presence of heavy oil would appear to prevent the engagement of opposing particles, and the heavier the oil the more effective is this interference. Perhaps the heat generated by the friction of the opposing surfaces in the case of a plug and ring when poorly lubricated has a much greater local expansive effect than is commonly suspected. It would seem that there is more involved in this problem than the mere question of mechanical fitting, and that it enters the domain of a complex physical problem.—EDITOR.]

CLOSING A KEYSEAT IN NICKEL STEEL SHAFTS.

Editor MACHINERY:

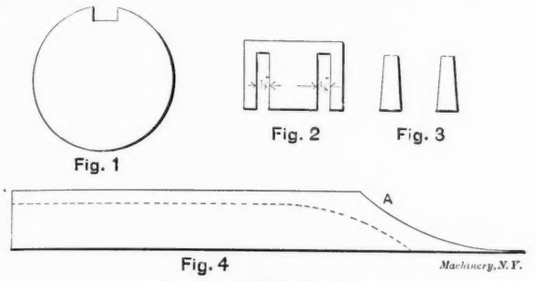
In all shops there are more or less mistakes made, and always will be. It is an established fact that the man who never makes mistakes is, generally, far less profitable to his employer than the man who is able to rectify his mistakes after he has made them, for it is "easy money" that the man who never makes mistakes is the man who has no "go-ahead" of his own. There is almost always some loophole for a man to pull himself out of trouble by if he uses his head, and I would like to show MACHINERY readers how we crawled out of a bad job, in hopes that it may help brother mechanics.

Having twelve large crankshafts to spline in a hurry, the man running the milling machine took his dimensions from another similar shaft, instead of looking up his drawing. This would have been all right had it not been that the sample shaft had the eccentric in place, while the shaft he was splining did not have the eccentric in place as yet. Therefore in allowing for the bearing, he neglected to allow for the eccentric, and consequently the spline was carried four inches into the bearing. As the shafts were nickel steel, you can imagine it was up to us to save them.

At first the foreman refused to allow pieces to be fitted in, and it looked dubious for a while. All kinds of suggestions were made, but he would not hear of tapping and riveting, as

is the common practice, for fear a rivet might work loose and ruin our reputation. On the other hand, I knew that if we lost those shafts, some one was going to lose his job, so after considerable persuasion I got permission to see what I could do, provided I took the responsibility and did not use any rivets. I really think the foreman thought he had me stuck, but as he was anxious to clear himself, he gave his consent.

Placing the shafts upon a vertical milling machine, I squared the end of the spline so that the piece could make a joint. Then with a small dovetail cutter, the sides of the spline were undercut the length of piece to be filled in. Care was taken that the top of the spline was not enlarged any. The end view, Fig. 1, shows how the spline looked after milling. Pieces were milled up from the same kind of stock to fit the top of the spline, which was ¾ inch wide. These pieces were milled to the gage and the end milled as shown at *A*, Fig. 4. Slots were milled 1-16 inch from each edge through the length of the piece to within ⅛ inch of the round in the key. These slots were ⅛ inch wide as shown in Fig. 2.



Saving a Spoiled Keyway.

Next I milled small wedge pieces as shown in Fig. 3, using the same cutter to taper the wedges that was used to taper the sides of the spline.

After inserting these two wedge pieces in the key, just hard enough to hold them, without spreading the sides, the piece was pressed into place by means of an arbor press. When the piece was pressed home, the wedge pieces spread the sides of the key to fill the spline in the shaft, making a fit which could never start out, and that without tapping and riveting. After slightly peening the edges of the joint to fill any cracks, the shaft and key were turned off in a lathe, and when ground it was impossible to detect it. Before carrying out this idea, a sample spline was tried, and after pressing home, it was attacked with hammer and chisel to see if it could be started. The piece had to be literally chipped out, and not until about two-thirds of the key was removed could the joint be started. TEXAS.

CLAMP WITH CHANGEABLE HEIGHT BLOCKING.

Editor MACHINERY:

Enclosed you will find a sketch which illustrates a changeable clamp device. The idea for this design I got in a machine shop where electric motors are built. To bore out the frames, they are set with their legs on the table of the boring machine and common clamps are used to fasten them up. Of course, as the motors are of different sizes, a whole pile of shimming and old nuts are required to put under the clamp. To avoid this I have the following idea: In a circle of 2½ inches diameter draw a regular pentagon with the base parallel to the diameter of the circumscribed circle as shown in Fig. 1. The left portion of the horizontal line from the center point to the outside of the pentagon is divided into two equal parts, the center being at *A*. From point *A* draw a line perpendicular to each side of the pentagon. By doing this we get five different positions of ½, 11-16, 1, 1 5-16, and 1½ inch. Suppose we have such a pentagon as described above made of cast-iron, say ¾ inch thick, and bring this in connection with a clamp, as shown in Figs. 2 and 3 by means of a rivet so that we can turn the pentagon. In this manner we get a handy tool very useful and practicable for lathes, planers, shapers, boring and milling machines, drill presses, etc. A pile of old nuts and blocks is not necessary any more. I suggest the use of a rivet for connection, to avoid separation. Fig. 3 shows the side view of the clamp. Three or four of these clamps will be enough for any kind of work in every shop. They are easy and cheap to manufacture and worth trying. MAX DEHNE.

Dayton, Ohio.

[A clamp on this principle was patented some years ago and, we believe, has been used to some extent.—EDITOR.]

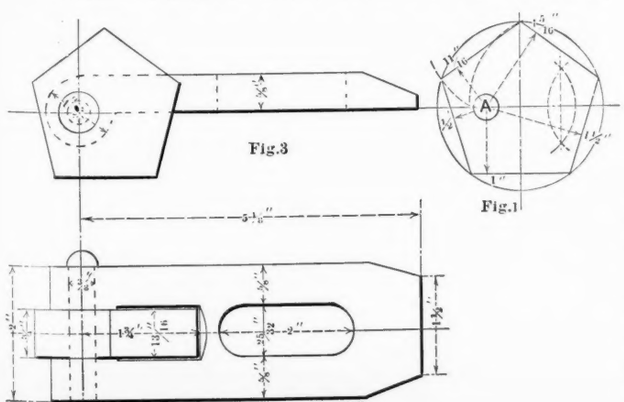
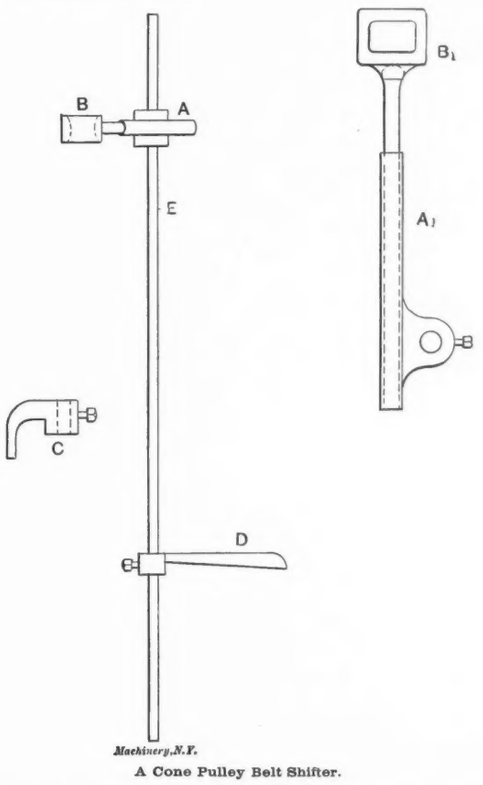


Fig.2
Clamp with Adjustable Blocking.

A HANDY BELT SHIFTER.

Editor MACHINERY:

The sketch shows a belt shifter which is as handy as anything going in its own line, for any belt running on cone pulleys, and especially for the lathe on account of the lathe belt having to be shifted often.



A Cone Pulley Belt Shifter.

This device is far ahead of having to use a stick, which is usually kept hanging on some convenient post. In using the stick the operator leaves his machine and goes where the shifting stick is kept, carries the stick to the machine, shifts the belt, then carries the stick back to its place. With the device here shown, the operator simply has to reach the projecting handle, give it a push and the belt slips off the upper cone and is quickly shifted to the corresponding cone on the lower pulley by hand. In shifting the belt the other way, the belt is first pushed off the lower pulley by hand and instantly shifted to the corresponding cone on the upper pulley by a pull of the handle on the shifter. The belt can be shifted

the whole length of the cone pulley by simply turning the handle far enough either way. Two stops fastened to the upright rod prevent the belt from being shifted off the pulley.

Referring to the sketch, *A* is a casting with a hole clear through the long part and a key-way milled through the side from near the base to within a short distance from the other end. The boss is bored to fit the rod *E* and a setscrew is fitted to hold it in place. *B* is the belt-guide, having a stem which fits loosely into *A*, and a pin driven in one side to act as a key to slide in the keyway in *A*. The belt-guide is riveted loosely to its stem to allow for sideplay. The reason for having the stem fit loosely in *A* is to allow adjustment to *B* for following the belt as it is shifted. At A_1 and B_1 is an enlarged sketch of *A* and *B* for a better understanding of it. *C* is a stop which is bored to fit the rod *E* and fitted with a setscrew for fastening. There are two pieces like *C*, placed one above the other on the rod *E* and fastened so that one or the other will bring up against the upper support for the rod *E*, when the belt is either on the largest or the smallest cone of the pulley. These stops of course are to prevent the belt slipping off the pulley. *D* is a handle bored to fit the rod *E* and fitted with a setscrew for fastening. The handle is either pulled or pushed to shift the belt. *E* is either a solid or hollow rod pivoted to the machine at the bottom and to the ceiling at the top in a manner best suited for the machine. It must be free to turn around as the belt shifts, and yet not too loose.

C. J. S.

TABLE OF MODULI OF CIRCULAR SECTIONS.

Editor MACHINERY:

In figuring the strength of pins and shafts for bending, it is necessary to know the modulus of the section. To facilitate such calculations, the accompanying table has been prepared. The formula $\pi\ d^3 \div 32$, the modulus of a circular section, was worked through for the following diameters: 1-32 inch to 6 inches, varying by 1-32 inch; 6 inches to 9 inches, varying by 1-16 inch; 9 inches to 11⅞ inches, varying by ⅛ inch, and the results were tabulated. "Vega," seven-place logarithm tables were used.

Example: It is desired to find the diameter of a projecting circular shaft to support a load of 5,000 pounds, 12 inches from the point of support. The allowable fiber stress for steel being assumed as 15,000 pounds, and using the formula $M = f\,s$, we have:

M = bending moment in inch-pounds.

s = modulus of section.

f = allowable fiber stress.

$$M = 5{,}000 \times 12 = 60{,}000 \text{ inch-pounds,}$$

$$60{,}000 = 15{,}000 \times s,$$

$$s = \frac{60{,}000}{15{,}000} = 4.$$

Looking in the table for the modulus 4, we find that the nearest number to 4 is 3.9887, which is the modulus of a section 3 7-16 inches in diameter. The size shaft to use would therefore be 3 7-16 inches.

O. E. THALEG.

So. Milwaukee, Wis.

MODULI OF CIRCULAR SECTIONS.

$$\frac{\pi\ D^3}{32} = .0982\ D^3$$

Dia.	0	1	2	3	4	5	6	7	8	9	10	11
0	0	.0982	.78560	2.6507	6.2848	12.2750	21.2112	33.682	50.279	71.918	98.20	130.70
1/32	.0000029	.10768	.82294	2.7350	6.4330	12.5066	.....	.....	.....	.....	.....	.....
1/16	.0000239	.11779	.86157	2.8205	6.5840	12.7411	21.8809	34.593	51.464	.....	.....	.....
3/32	.0000809	.12554	.90127	2.9077	6.7360	12.9775	.....	.....	.....	.....	.....	.....
1/8	.0001918	.13981	.94229	2.9968	6.8910	13.2188	22.5647	35.519	52.672	74.612	101.92	135.21
5/32	.000374	.15178	.98441	3.0875	7.0502	13.4621	.....	.....	.....	.....	.....	.....
3/16	.000647	.16069	1.0279	3.1803	7.2106	13.7083	23.2625	36.462	53.897	.....	.....	.....
7/32	.001028	.17775	1.0725	3.2745	7.3730	13.9576	.....	.....	.....	.....	.....	.....
1/4	.001534	.19179	1.1185	3.3710	7.5384	14.2098	23.974	37.422	55.141	77.720	105.50	139.82
9/32	.002184	.20652	1.1657	3.4690	7.7056	14.4651	.....	.....	.....	.....	.....	.....
5/16	.002996	.22202	1.2143	3.5692	7.8758	14.7234	24.701	38.398	56.403	.....	.....	.....
11/32	.003988	.23824	1.2642	3.6711	8.0481	14.9847	.....	.....	.....	.....	.....	.....
3/8	.005178	.25528	1.3165	3.7751	8.2233	15.2492	25.442	39.391	57.685	80.914	109.65	144.53
13/32	.006584	.27305	1.3680	3.8808	8.4198	15.5167	.....	.....	.....	.....	.....	.....
7/16	.008223	.29477	1.4221	3.9887	8.5808	15.7873	26.197	40.401	58.987	.....	.....	.....
15/32	.0101140	.31110	1.4774	4.0984	8.7630	16.0610	.....	.....	.....	.....	.....	.....
1/2	.01227	.33142	1.5343	4.2103	8.9485	16.3380	26.968	41.428	60.307	84.194	113.67	149.35
17/32	.014734	.35259	1.5955	4.3200	9.1359	16.6181	.....	.....	.....	.....	.....	.....
9/16	.017477	.37460	1.6523	4.3399	9.2837	16.9017	27.753	42.472	61.647	.....	.....	.....
19/32	.020555	.39742	1.7134	4.5576	9.5191	17.1878	.....	.....	.....	.....	.....	.....
5/8	.023975	.42137	1.7762	4.6777	9.7151	17.4774	28.554	43.552	63.021	87.561	117.78	154.28
21/32	.027766	.44611	1.8403	4.7996	9.9130	17.7295	.....	.....	.....	.....	.....	.....
11/16	.031920	.47189	1.9061	4.9239	10.114	18.0665	29.370	44.613	64.386	.....	.....	.....
23/32	.036462	.49856	1.9733	5.0499	10.313	18.3660	.....	.....	.....	.....	.....	.....
3/4	.041428	.52629	2.0422	5.1785	10.524	18.6683	30.201	45.710	65.786	91.017	121.99	159.30
25/32	.046826	.55494	2.1125	5.3088	10.733	18.9757	.....	.....	.....	.....	.....	.....
13/16	.052672	.58471	2.2355	5.4418	10.945	19.2841	30.976	46.815	67.206	.....	.....	.....
27/32	.058986	.61544	2.2582	5.5765	11.159	19.5968	.....	.....	.....	.....	.....	.....
7/8	.065786	.64731	2.3346	5.7138	11.3771	19.9129	31.910	47.958	68.646	94.563	126.29	164 44
29/32	.073089	.68017	2.4104	5.8529	11.5973	20.2323	.....	.....	.....	.....	.....	.....
15/16	.080914	.71422	2.4891	5.9948	11.8205	20.5552	32.788	49.109	70.106	.....	.....	.....
31/32	.089278	.74929	2.5693	6.1384	12.0462	20.8815	.....	.....	.....	.....	.....	.....

MECHANICAL ADVERTISING.

Editor MACHINERY:

The man who said that the superlative degree was invented for advertisers spoke somewhere near the truth. While no doubt in the best advertisements moderation is practised, many appear in which there is so much straining after "going one better than the other fellow" as to become absolutely impossible. Indeed, it often becomes a question whether the "ad" shows the writer's ignorance or his knowledge of other people's. For instance, a British copper wire manufacturer announces that his cable has 100 per cent conductivity. Now I do not know much about things electrical, but a cable is, theoretically, a machine and you cannot get out of it as much as you put in. Another English firm sells "frictionless engine packing." The quotation marks are mine. That these things are absurd ought to be obvious to anyone in the position to have any say in the matter of buying them. Why, then, are they made a point of in the advertisements?

To successfully advertise is no easy matter. The "ad" should be so compiled that men interested in such things look for it, one might almost say, habitually. The attracting of the eye is generally more, I take it, the printer's work than the compiler's of the ad; a big headline, or perhaps the reverse, a small amount of matter in a large space, a taking photo, all tend to attract. To hold the attention when once attracted the matter must be interesting. In this the me-

chanical advertiser has some little advantage; a man advertising, say, a patent medicine has to create an interest in his wares, but the compiler of a mechanical "ad" has generally the interest there—he needs only to stimulate it.

In conclusion, if I were writing a book on advertising I think I should lay down as a postulate that the art of "ad" writing does not consist of packing as many superlatives to the square inch as is possible. McANIC.

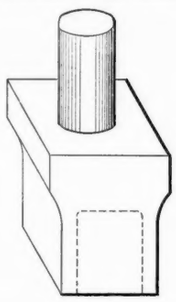
AN ADJUSTABLE DIE—AN ANGLE CUT-OFF DIE

J. L. Lucas.

Editor MACHINERY:

We had a large number of contact pieces for electrical work to be cut from pure silver, and it was important not to waste the stock more than was absolutely necessary. At first we cut them out with hand shears, but the number wanted was so large, and the process was so slow, that a die like the sketch was made for the work. The operation of the same was so satisfactory that I think it will be interesting to those engaged in electrical work. It is so simple, an explanation is hardly necessary.

The gage *A* is set to the width you want and the stop *B* is set to the length of the piece wanted, and then in feeding, the work is held against the back gage and pressed up to the stop. The sizes that can be cut are only limited by the size of the die. The punch is cut out as shown to allow the stop to project up so the silver will strike it. Both punch and dies are hardened in the usual manner.



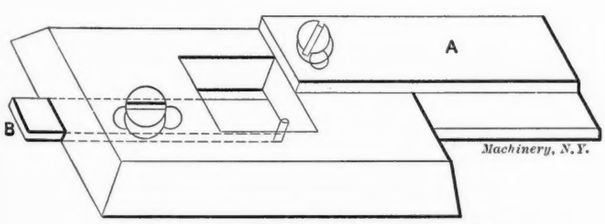


Fig. 1. A Die for Cutting Silver Blanks without Waste.

In cutting off angle iron for automobile frames, or other work of that nature, it is desirable to have both ends of the work left square, which an ordinary angle iron cutter will not do.

The punch and die shown were made to overcome this defect, and they do it very satisfactorily. They leave a nice true surface on both ends of the piece cut off. There is only one objection to the use of the die, and that is, there is one-quarter inch waste on every cut, or whatever may be the thickness of the punch.

The punch shown is made at an angle of forty-five degrees, so as to give a shearing cut.

JAMES L. LUCAS was born at North Carver, Mass., 1855. He served an apprenticeship with E. T. Jenks, Middleboro, Mass., and has since worked for the Mason Machine Works, General Electric Co., Domestic Needle Co., and Mossberg & Granville Co., in the various capacities of toolmaker, foreman, draftsman, salesman and superintendent. His specialty is dies and press tools, and he has written a book on the subject, called "Dies and Die Making," which has had a considerable sale. Mr. Lucas has contributed frequently to the trade papers, generally under the nom-de-plume of "A. P. Press."

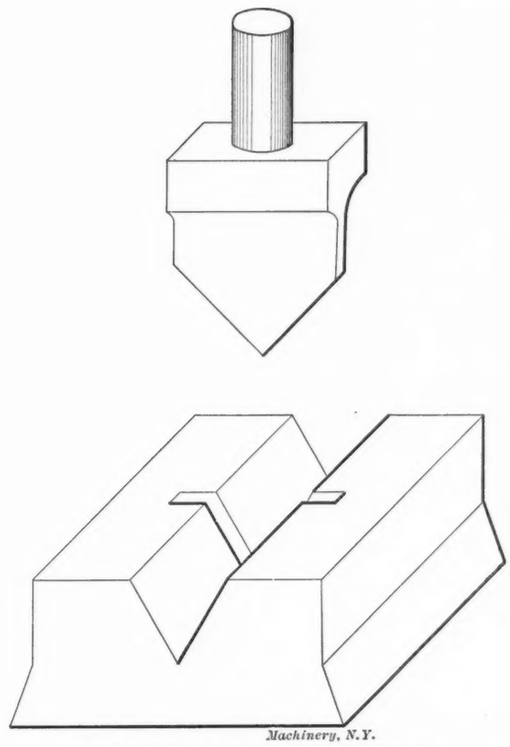


Fig. 2. Punch and Die for Cutting Angle Iron.

The time spent on the die in making, was twenty-two hours, and it has paid for itself many times over. It takes in all sizes from one to three inches, and works equally well when there is a long and short leg to the iron. J. L. LUCAS.

Bridgeport, Conn.

SHELL COUNTERBORING TOOL.

Editor MACHINERY:

The piece of work shown in Fig. 2 is the clutch hub of an automobile, and it is a bronze casting. A helical spring, made of 5-16 inch square stock, is required to work smoothly in the recess *A*.

It was first tried to cast this recess, but there always were fins left at the bottom that had to be chipped out, and the sides were so rough that the spring would not work smoothly.

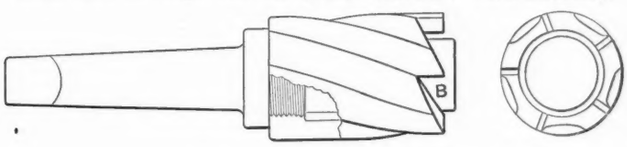


Fig. 1. Shell Counterboring Tool.

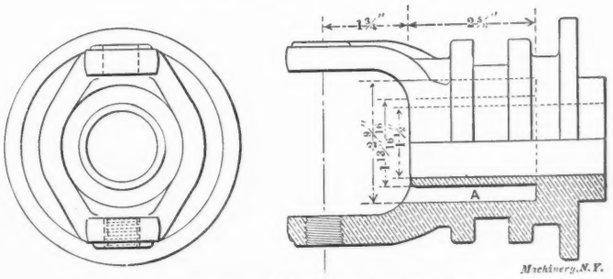


Fig. 2. Showing Cut taken by Shell Counterbore.

It was then attempted to clean out the recess in a lathe, using a tool similar to a cut-off tool, but having a blade bent in an arc. The tool was necessarily long and springy. The lathe hand succeeded in doing about twenty pieces before it broke. Another tool was made, which lasted about as long as the first. Then the tool shown in Fig. 1 was constructed, and has proven quite successful.

It consists of two pieces, being a tool steel shell, screwed upon a machinery steel shank. The shank is tapered to fit the tailstock of a lathe, and extends through the shell so as

to form a pilot, which enters the 1½ inch reamed hole shown in Fig. 2. The shell has five cutting teeth, and five spiral flutes to carry away the chips. The tops of the flutes are relieved like the flutes of a twist drill. The inside diameter of the shell is 1-64 inch larger at the back than at the cutting edge, and the outside diameter is 0.005 inch less at the back than at the front.

The work at present is being done in an engine lathe, the casting being chucked, the 1½ inch hole bored and reamed, and the recess bored with the above tool. The work is then taken from the chuck and finished on a mandrel.

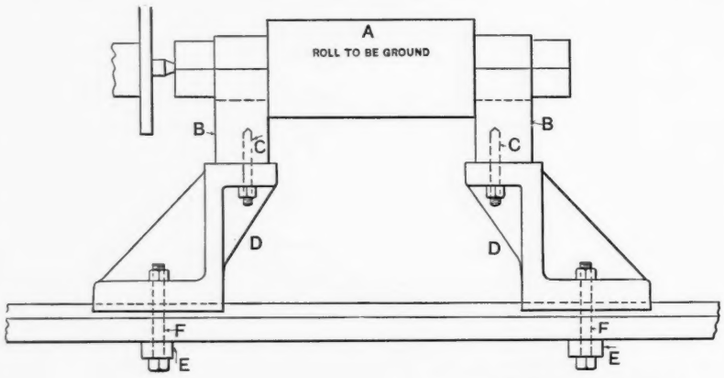
Indianapolis, Ind. OTTO L. LEWIS.

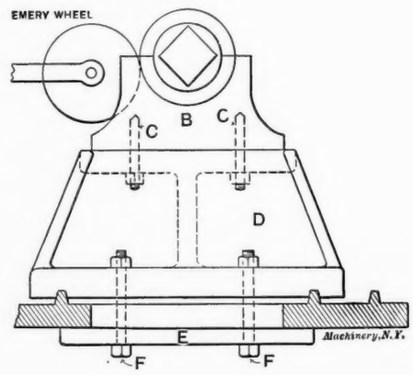
GRINDING EIGHT-INCH FINISHING ROLLS.

Editor MACHINERY:

Permit me to present a method of grinding eight-inch steel rolls by means of a lathe fixture which has been found to be both effective and successful. In the accompanying sketch, *A* is the roll to be ground; *B* represents two cast iron bearing blocks with two steel studs *C* driven into each and threaded on the extending part. The cast-iron brackets *D* are offset in order to allow full travel for the carriage on the lathe bed. In order to dispense with vibration, these castings are made particularly stiff and strong, the offset parts being heavy and joined by stout stiffening ribs. On the top of each bracket where the bearing block *B* rests are two slotted holes into which studs *C* fit, being fastened on the under side of the shelf by means of nuts. These holes give all necessary adjustment in setting the rolls. The brackets are gripped on the lathe-bed by clamps consisting of wrought-iron straps *E* with two bolts *F* through each strap.

This fixture is placed on the inside ways of the lathe bed while the carriage travels on the outside ways. The emery wheel is driven by a motor attached to it, both the motor and wheel being placed in the toolpost of the lathe. The cooling water runs down over the top of the roll from a tube connected





Rig for Grinding Rolls.

to any convenient receptacle. The wires which bring the supply of current are connected to the motor on the toolpost, the power for the lathe being a separate consideration. This grinding arrangement, simple in detail and easy to set up and adjust, insures full travel for the grinding wheel over the surface being finished and moreover performs the work rapidly and accurately. W. T. M.

Providence, R. I.

METHOD OF MACHINING BUSHES.

Editor MACHINERY:

The following is a method which we have found very satisfactory for turning out brass and babbitt bushes in large quantities at a minimum cost. The quality of the work produced is very accurate.

We have a large number of bushes to make, a good many of them of the same diameter, but of different lengths, so we make all our castings about 10 or 12 inches long. To save truing up, we have found it economical to make a special faceplate with a shoulder on it, as shown at *D*, Fig. 1. To this we clamp the cast-iron center, Fig. 2, and in the place of the tail center we put another style of center, Fig. 3. This tail center is made of cast iron and the spindle of tool steel, hardened

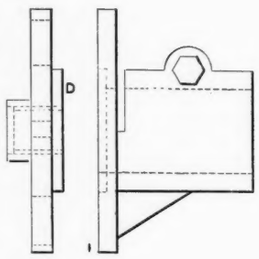


Fig. 1.

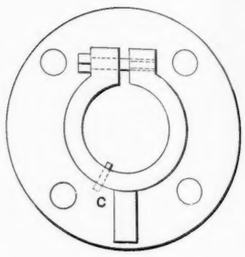


Fig. 4.

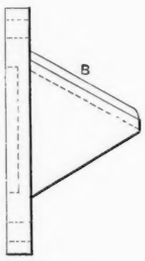


Fig. 2.

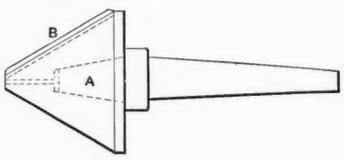


Fig. 3.

Machinery,N.Y.

Lathe Attachments for Machining Bushings.

and ground. The part *A*, on which the center turns, is made taper to insure it remaining tight and true when it begins to wear.

The core of the bush casting is then centered at each end with a large center reamer, and a small slot chipped in each end to clear the keys *B B*. A 1¼-inch key projecting ⅛ inch has been found sufficient to drive the heaviest cut, and a much smaller key will do all right in Fig. 3.

When the bushes have been turned to size on these centers we remove the head center and in its place clamp the chuck, Fig. 4. This chuck was bored the exact size of the bush on this same faceplate with the clamp bolt tightened slightly, so that when the bolt is loosened the bush slips in easily, and when tightened again is absolutely true. The slot has been found necessary to make this chuck clamp the work parallel with the center line. The bushes are then cut off and bored out in this chuck. In case the bushes have to be correct for length, we put a pin, *C*, in the side of the chuck for a stop, and face the end of the chuck the correct distance from the pin. All that is then necessary is to slip the bush in against the pin and set the tool by the end of the chuck for the finishing cut.

We have found this method to be a time-saver where there are a large number of bushes to make.

Amherst, N. S. S. J. FISHER.

* * *

The Pennsylvania Railroad Co. has placed the largest equipment order in this country. The order consists of 500 locomotives and 15,000 freight cars; 250 of the locomotives are to be built by the Baldwin Locomotive Works, and 250 by the company's shops at Juniata, Pa. The value of the locomotive order is $7,250,000 and of the cars $15,000,000.

SHOP KINKS.

A DEPARTMENT OF PRACTICAL IDEAS FOR THE SHOP.

Contributions of kinks, devices and methods of doing work are solicited for this column. Write on one side of the paper only and send sketches when necessary.

TO ANNEAL HIGH-SPEEED STEEL.

High-speed steels may be very thoroughly annealed by packing in a cast-iron box in gray cast-iron chips. The steel should be surrounded by an inch or two of the chips. Then heat slowly in the gas furnace (or a blacksmith's forge will do) to a cherry red. Hold the heat there for from one to three hours, according to the size of the pieces to be annealed, cover the box well and allow to cool slowly—in the gas furnace if possible.

I. W. ANTANO.

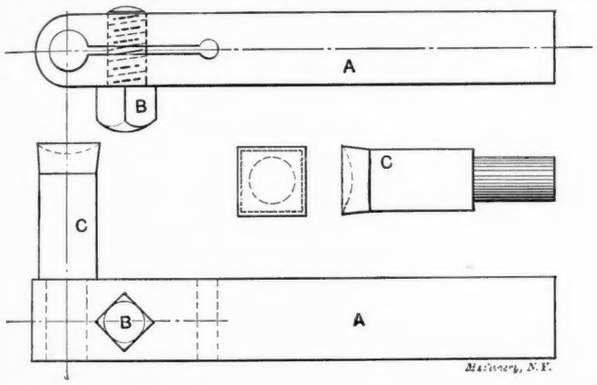
TO HARDEN SMALL JIG BUSHINGS.

To harden large quantities of small jig bushings without danger of cracking under the head while hardening or while driving them home, proceed as follows: Put one gallon of fish oil in a suitable metal bucket and place in a larger bucket of cold water. The bushings, strung about six on a wire, are heated in a small blow torch fire to a light red heat and are then quickly plunged into the oil, and kept moving around until cold. The hardness will depend upon the degree of heat given, and this can be so regulated that it will not be necessary to polish and draw bushings after hardening.

New York. H. J. BACHMANN.

SHAPER TOOL FOR SQUARE AND OBLONG HOLES.

I am sending you a sketch of a shaper tool for working out square and oblong holes at one setting. As shown at *C*, the tool cuts on the end, has relief at all four sides, and is cupped out at the center to allow the chip to curl. The other end is



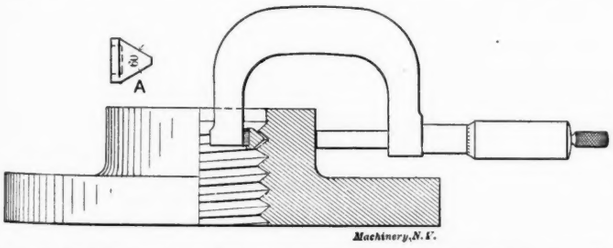
turned down to fit the hole in the shank, *A*. This shank is split as shown in the top view, so a wrench applied to the screw, *B*, will hold the tool firmly in place. When in use, the clapper of the tool head is wedged down.

St. Paul, Minn. ARTHUR MUNCH.

USE OF MICROMETER FOR INTERNAL THREAD CUTTING.

Editor MACHINERY:

The accompanying sketch illustrates a means of determining the threaded size of duplicate work. The plate shown is intended for a lathe chuck. The outside diameter of the hub



on this plate is turned to the same size as the hubs on small faceplates which are furnished with all new lathes. The threaded size is then taken and transferred with a micrometer, over the anvil of which is fitted a 60-degree point as shown enlarged at *A*. In connection with a graduated cross-feed screw this greatly facilitates the work over the usual cut-and-try method.

CHARLES SHERMAN.

Detroit, Mich.

LINING UP LATHE CENTERS.

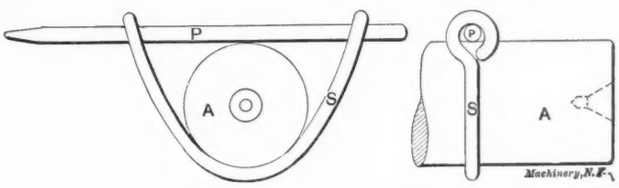
To line up the lathe centers to turn straight, turn up about an inch on the end of a small piece, say about 2 feet long, being careful to make it perfectly round. Put a tool in the tool-post with the blunt end of the tool next to the work and have the end of the tool free from burrs. Turn the bar end for end, having the turned end at the live center. Now run the tool up against the finished part of the bar, taking care to come in contact with the piece carefully. Make a light scratch with a scriber on both the cross-slide handle and the sleeve in which the cross-slide screw runs. Next draw back the tool from the work and turn the piece around so the turned part is at the tail center, then run the carriage up and feed the tool up and adjust the screws in the tail-stock until the marks made by the scriber on the cross-slide handle and sleeve come in line, with the tool lightly against the work.

Detroit, Mich. C. J. SHAW.

SPRING FASTENER FOR LATHE POINTER.

Editor MACHINERY:

The accompanying sketch of a feeler or pointer shows one that is used for setting up work in a lathe where the work has a hole through it, and is to be trued up by the hole. An arbor *A* is placed between the lathe centers, and the spring *S*, which is a straight piece of spring wire with an eye formed at each end, is sprung around the arbor and the pointer *P* passed through the eyes and over the arbor, making tension



enough to hold the pointer in place. The arbor is rotated by hand and carries the pointer around with it to test the accuracy of the setting. This device is of the most service for heavy and awkward-shaped pieces, as they can be set and fastened without starting the lathe. It is also very handy for setting on the lathe carriage cylinders, bushings, etc., that are to be bored with a boring bar. A very light arbor can be used together with a short spring and pointer when the hole in the work is small. With the end of the pointer bent hoop-shaped it can be used for setting to the outside of a hub.

Watervliet, N. Y. M. H. BALL.

* * *

The publication of the Rand Drill Co., known as *Air Power*, issued under the direction of Mr. P. F. Kobbe, is discontinued with the October issue. Although it had been published quarterly for only one year it had come to be recognized as one of the best class of house-organ publications. The combination of the Rand Drill Co. with the Ingersoll-Sergeant Co. under the name of Ingersoll-Rand Co. made the publication of *Air Power* unnecessary as the older publication, *Compressed Air*, issued by the Ingersoll-Sergeant Co., will be continued.

* * *

A correspondent sends us an advertisement of a washing machine which he has clipped from a farm paper. It contains an interesting description of the manner in which the springs and ball bearings in the aforesaid machine do practically all the hard work that usually falls to the housewife's lot on Monday morning. To quote from the advertisement, "In using this machine a *little* help is needed from you each time, but the springs and the ball bearings do nearly all of the hard work." We cannot but wonder why the inventor did not add a few more springs and ball bearings so that they might do all of the work, perhaps even to the extent of making a brake a necessary part of the machine to prevent the mechanism from running away.

SHOP RECEIPTS AND FORMULAS.

This page will be used for the publication of shop receipts which the contributors know from experience to be practicable. Nearly all readers of MACHINERY can add something, and it is desired that they use this page as a medium for exchanging useful formulas, thus putting them on record for the general benefit of the craft. All published receipts are paid for.

102. TO IMITATE CASEHARDENING.

Occasionally it becomes necessary to darken polished or ground parts to imitate casehardening; in order to accomplish this result use this mixture: 1 part nitric acid and 20 parts water. Immerse the article to be treated about 20 seconds, then rinse with clear water. A splendid result can be accomplished by following the above instructions. HARRY ASH.

Chicago, Ill.

103. BRIGHT DIP FOR BRASS, COPPER AND BRONZE.

A bright dip for brass, copper and bronze may be produced as follows: Make a solution of 100 parts by weight of nitric acid, 50 parts sulphuric acid, 1 part soot, and 1 part salt. The salt and soot make the dip work smoothly. The article should be dipped in this solution, well washed, and dried in sawdust to prevent streaking. S. H. SWEET.

Bridgeport, Conn.

104. PACKING FOR GAS ENGINE CYLINDERS AND PIPE CARRYING GASOLINE VAPOR.

To prepare packing for joints in pipes, etc., carrying gasoline vapor, mix a quantity of graphite and kerosene to a thick paste and apply the paste to both sides of sheet asbestos. When dry the packing may be cut to the shape desired. The graphite helps the asbestos make intimate contact with the iron and thus maintain a tight joint continuously at high temperature for an indefinite time.

New York. H. J. BACHMANN.

105. BRASSING IRON.

Iron ornaments are covered with copper or brass, by properly preparing the surface so as to remove all organic matter which would prevent adhesion, and then plunging them into melted brass, or copper. A thin coating is thus spread over the iron, and it admits of being polished, or burnished. The better the article is finished and cleaned before dipping, the better will be the final result. R. B. CASEY,

Schenectady, N. Y.

106. COMPOUND FOR CLEANING THE HANDS.

To loosen the oil and grease, the hands should first be scrubbed with a stiff brush dipped in kerosene, and then they should be wiped dry with waste. Take a five-cent box of any kind of soap powder (I prefer Soapine, because it lathers freely) add to it an equal quantity of sawdust and half the quantity of white sand. Mix thoroughly and rub over the wet hands in the form of a paste. This entire mess will rinse off in any kind of hard or soft, hot or cold water. Hands washed in this manner twice a day will be free from grime and clean all over. H. J. BACHMANN.

New York.

107. CEMENT FOR STICKING EMERY PAPER ON SMALL GRINDER DISKS.

Melt together 5 parts paraffine, 4 parts bees' wax and 1 part rosin. When cold, cut into blocks, and apply evenly on the revolving disk until it has a thin coat over its entire surface. The emery paper should then be pressed on the disk while it is still revolving, thereby slightly heating both the disk and the paper, and causing the cement to spread in a thin layer all around the disk. The belt should then be shifted onto the loose pulley, so that the paper may be pressed closely to the disk. The corners may then be trimmed off with an old file. It requires a little practice to perform the job successfully, but the method is much superior to removing the disk and gluing the emery paper on in a press. The worn-out paper can be more easily removed, it being only necessary to wait until the disk is cool, when, by taking hold of one portion of the paper, it may be ripped right off.

New York. H. J. BACHMANN.

108. CEMENTS FOR IRON JOINTS.

The following are cements used to make the joints of machinery air and water tight:

1. Mix ground white lead with one-fourth its weight of red lead.

2. Mix equal parts of red lead and white lead, in powdered form, with enough boiled linseed oil to make the whole a soft, putty-like mass.

3. To 50 pounds of borings, preferably cast iron, which have been pounded and sifted, add one pound of sal-ammoniac. Mix with water when ready to use.

4. Boiled linseed oil and red lead mixed to the consistency required. A small quantity of litharge improves the cement for many purposes.

5. Cast iron borings, 4 pounds, dried potter's clay 1 pound, powdered potsherds (broken crockery) 1 pound. Make into a paste with salt and water.

I have used all of these cements and find them satisfactory No. 3 is used largely for filling cracks in boilers, etc., and No. 5 is excellent for outdoor iron work, water tanks, etc.

New Britain, Conn. F. L. ENGEL.

109. TO TREAT INFLAMED EYES—CARE OF THE EYES.

The treatment of an inflamed eye is a matter of some moment in a machine shop and too much care cannot be taken to treat such cases scientifically. You have only two eyes—unless you are a foreman and then you are supposed to be a full-fledged pineapple, as far as eyes are concerned. A splendid remedy for an inflamed optic is a weak solution of powdered borax water—either warm or cold—applied by rubbing it in the eye with a cloth, or dropping it in. It is very soothing and will drive the soreness and inflammation out and leave the eye in a better condition than it was before it was irritated. The proper proportion is a spoonful of powdered borax to a glass of water. A mechanic should always bear in mind that the loss of an eye may drive him to selling shoe-strings. A pair of plain eyeglasses will protect the eyes from chips or emery, and borax water is good for tired eyes too—the kind of eyes you have when working too much overtime.

Another eye kink is to get a round looking glass about 3 inches in diameter, and on the back of it near the center attach a cloth band or strap. I made one with a ball-and-socket joint. Now this strap is made to fit easily around the head, the glass resting against the forehead. The function of this glass is to reflect the light from a distant window on to the work. It is a very satisfactory rig to wear when filing to a line or working to a line on a machine. The finer the glass the better is the focus of light reflected. By using a ball-and-socket joint the glass can be instantly adjusted to throw the light on the point you wish to see. Every diemaker ought to have one. CARROLL ASHLEY.

Rochester, N. Y.

* * *

The place that scraping fills in the fitting and finish of modern machine tools is not, in the best practice, that of making the working surfaces more accurate, but of giving them smoothness and hardness that approximates the well-known condition of a worn cast iron surface. The mistaken estimate of scraping was that it is done to enhance accuracy. This was true, and is still true to a certain extent for small surfaces, but it is largely untrue when it comes to the matter of scraping on housing faces and other parts which have a great superficial area. It is manifestly impractical for a fitter to scrape down a large area any perceptible amount. The planer hand must be held to strict accountability, and if the work produced by his machine does not show truth within a very few thousandths it must be replaned, rather than scraped. And even if it were practicable to reduce large surfaces, the scraping process is more or less unreliable for it is dependent upon a system of straight-edges and surface plates which require great care if they maintain their truth, and in the large sizes are very difficult to handle without distortion. The place to secure accuracy is on the planer; the scraping should be done to improve the surface qualities and thus reduce wear and friction.

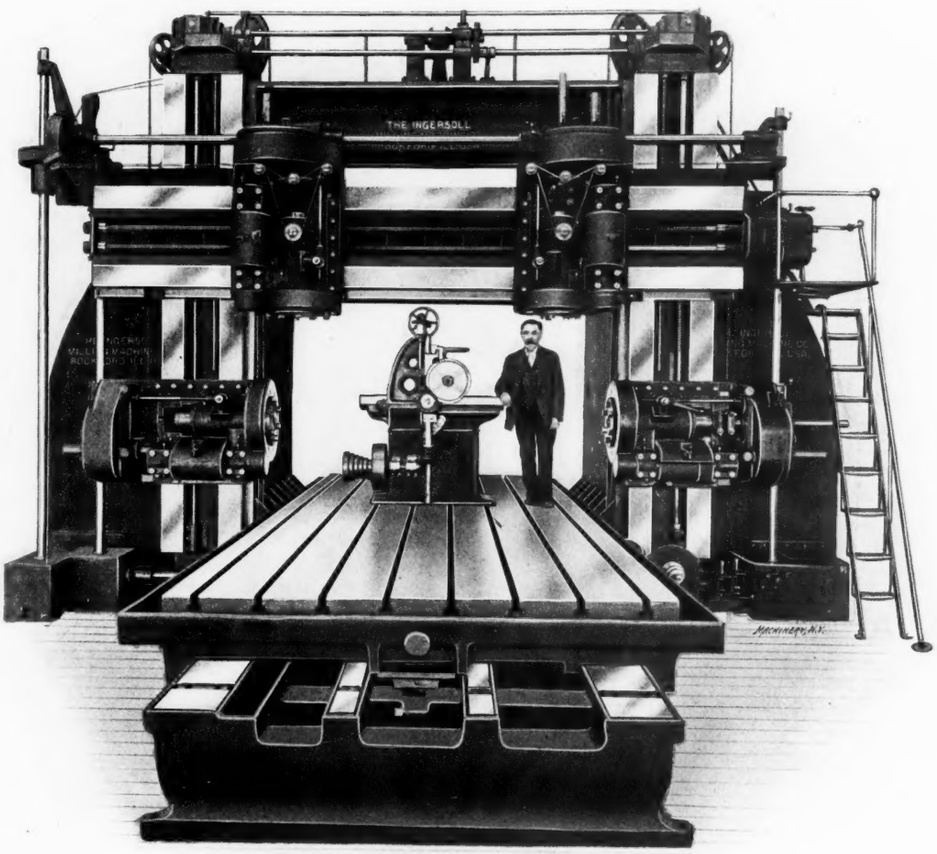
MACHINERY AND TOOLS.

A MONTHLY RECORD OF NEW APPLIANCES FOR THE SHOP.

HEAVY FOUR-SPINDLE MILLING MACHINE.

The Ingersoll Milling Machine Co., of Rockford, Ill., have recently built the milling machine shown in the accompanying cut. It is of unusually large size and was built for the General Electric Co., Schenectady, N. Y. It is possible to get some notion of its capacity by comparing it with the smaller machine, one of the first lot built by the firm. This new machine is of the planer type with two vertical heads on the cross rail and two horizontal heads on the housings. The saddles are all of the same type. The large face mills which are used, are mounted on face plates driven by gear teeth cut in their periphery. The cutters in each saddle have independent adjustment, that is, the vertical saddles have adjustment vertically and the horizontal saddles have adjustment horizontally, 8 inches in each case. The right hand vertical and horizontal saddles each carry auxiliary spindles for small face mills. The two spindles on the cross rail have independent reversible cross feed and the right hand horizontal spindle has independent up and down feed. Both horizontal saddles, the cross rail, and the two vertical spindles are raised and lowered by power and are counterbalanced. The machine is driven by four motors, one of 50 horse power, which drives the two horizontal heads, one of 50 horse power for the two vertical heads, one of 15 horse power for the table feed, and one of 5 horse power for raising and lowering cross-rails; these are all General Electric variable speed motors. The spindle drives and feeds are thus independent of each other.

The housings are braced by heavy buttresses to take the side thrust necessary to carry a big face mill along the crossrail, under the pressure of a heavy cut. This entails considerable strain, especially if the cross rail is near the top of the housing. The machine will take a casting 10 feet high and 10 feet wide. The table is 20 feet long within the pockets, 108 inches wide over all, and has four bearings, two near the outside edge of the table and two supplementary ones near the center. The shipping weight of the machine was 300,000 pounds.



Ingersoll Four-spindle Milling Machine. The small Milling Machine on the Platen is the first one Built by the Company, in 1887.

TOURIST AUTO KIT.

The kit of tools shown in the cut is designed, as its name implies, for the use of automobilists in repairing breakdowns on the road. Most of the tools are plainly shown in the drawing, yet there may be a few whose use is not apparent.

No. 1 is a roll of emery cloth, containing three different grades.

No. 3 is a soldering stick.

Tourist Auto Kit.

No. 20 is a chain holder for connecting or disconnecting chains while on the road.

No. 26 is a roll of adhesive tape.

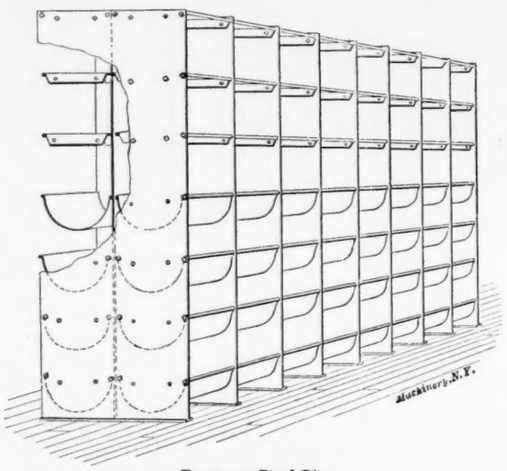
The three nickel-plated cases, Nos. 29, 30 and 31, contain a gasoline hydrometer, a supply of emery for grinding valve seats, and an assortment of cotter pins of various sizes.

No. 32 is a cotter pin extractor.

The whole outfit rolls up into a compact canvas case; it is sold by Hammacher, Schlemmer & Co., New York, N. Y.

THE ECONOMY STEEL BIN.

This bin, of which a line engraving is shown below, is built by the Davis Mfg. Co., of Milwaukee, Wis. The frame is made of 16-gage and the bins of 26-gage sheet steel securely bolted and braced together. The bins are furnished with either level or pocket bottoms. Those with the level bottoms are intended for stock put up in packages or boxes, while the



Economy Steel Bin.

pocket bottom bins are intended for such articles as nuts, bolts, screws, pipe fittings, etc. They are made in single sections to be placed against the wall or in double sections placed back to back. Every bin is accessible. They are fireproof and can be supplied in any length or height. They are shipped knocked down, packed in crates.

"GARDNER'S IMPROVED" DISK GRINDER.

The accompanying half-tone shows a new disk grinder designed by Mr. F. N. Gardner, and built by the Gardner Machine Co., Beloit, Wis. The thrust of the spindle of this machine is taken at the right-hand box on hardened steel col-

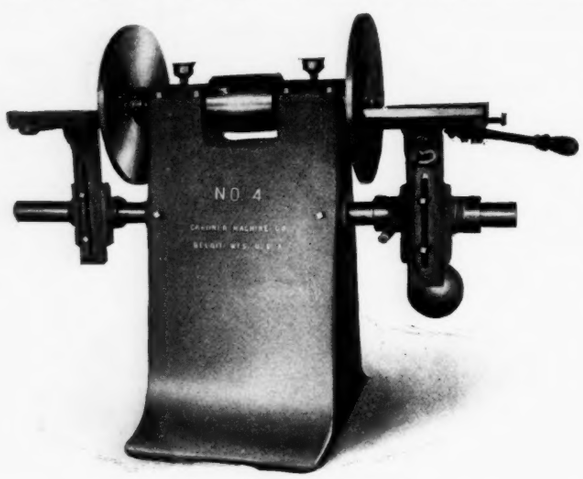


Fig. 1. Gardner Disk Grinder.

lars 4 inches in diameter, giving about 9 square inches of bearing surfaces at each end. These collars bear against cast iron surfaces plugged with babbitt. The design is such that if the spindle heats, the thrust bearing will loosen instead of sticking, as is the case with the usual arrangement. The construction of the work tables is sufficiently well shown in the half-tone. On the left hand table the top may be tilted to 45 degrees and the gage strip on the working surface is capable of swivelling an equal amount. The table at the right has a lever feed. By its use ample pressure of work against the feed can be obtained on all sizes and classes of work. The table is mounted on a gibbed slide which is protected from the dust by sheet metal guards on the under side. It is provided with T-slots for half-inch bolts. The movement of the table is limited by a fine threaded stop-screw at the back and an adjustable counterweight allows a correct balance to be maintained. This latter table is an extra attachment.

Fig. 2. Press for Cementing Cloth on Disks.

The wheels for this machine are 20 inches diameter by ¾ inch thick. The spindle is 2 inches in diameter with bearings 8 inches long. It is driven by a 7-inch pulley with an 8½-inch face.

Fig. 2 shows a process for cementing emery cloth to the disks. Pressure is obtained by a hand lever and screw. By means of the lifting attachment, heavy wheels can be handled with ease. The lifting arms can be used for only one size of wheel, but extra sets of arms with screw attached are furnished for any size desired up to 23 inches in diameter. By using this arrangement the wheels may be handled in a damp condition without the risk of displacing the cloth or the exertion of lifting, which are troublesome when the disks are shifted by hand.

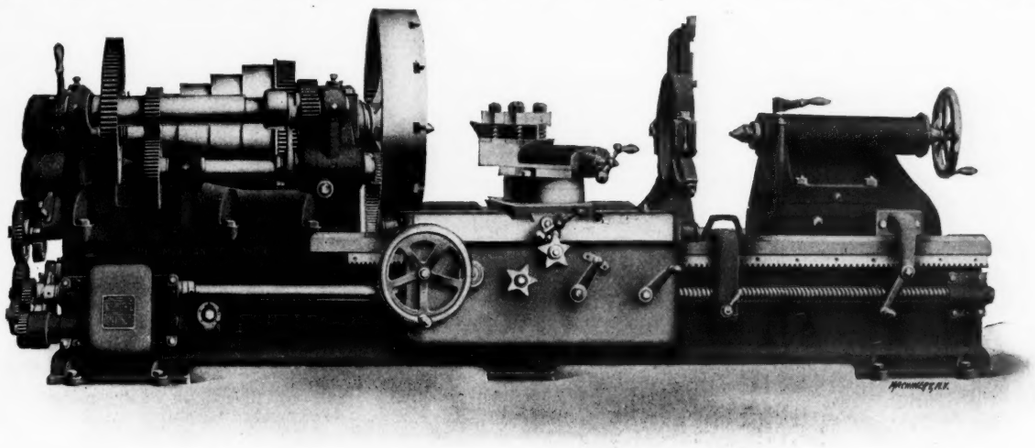
THE "STANDARD GUIDE" STEAM HAMMER.

The David Bell Engineering Works, Buffalo, N. Y., which makes an exclusive specialty of the building of steam hammers, has brought out a new line of single frame machines which is called the "Standard Guide." The main frame of the hammer is cast without cores, excepting, of course, in the column, which is of box section. This gives uniformity of thickness and shape to the casting. The bed plate is solid with the frame, with heavy ribs below the floor, and reinforced around the die block. This pattern of hammer takes its name from the fact that the main casting to which the lower ends of the slides are bolted is made stronger and heavier than usual. This overcomes a weak point in this type of hammer, and greatly strengthens the place where the most strain comes. They are regularly made in seven sizes, ranging from 350 to 2,000 pounds falling weight.

FORTY-TWO INCH "AMERICAN" LATHE.

As the half-tone shows, this lathe is of massive design throughout. It is one of a new line which the builders, the American Tool Works Co., Cincinnati, Ohio, have built to meet the requirements of modern practice. The feed is driven through a quick change gear mechanism which provides 32 changes for feeding and thread cutting, the range of threads being from one thread in 4 inches to 16 threads per inch, including 11½ pipe thread. The feed range is from 6.4 to 92 cuts per inch. The device is operated while the machine is

running, if necessary, by a revolving nut seen at the right of the gear box beneath the head, which moves a sliding key engaging two opposite gears, each being one of a cone of gears which is encased in the gear box. The feed or screw pitches thus obtained are multiplied by the compound gears on the quadrant at the end of the head, it being necessary to change one gear only on the quadrant for each additional thread. This arrangement gives flexibility to the screw cutting mechanism, making it possible to cut an unlimited number of sizes of threads or worms, either finer or coarser than the range indicated above. An index plate is provided to assist in obtaining the desired feed or pitch.

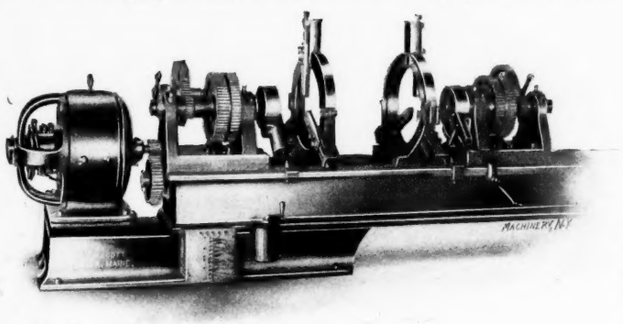


American Tool Works Co.'s Forty-two inch Lathe.

The head stock is triple geared with bearings of phosphor bronze, and with the five-stop step cone divides the feed range into 15 steps in geometrical progression. The feed may be reversed in the apron, a feature which is valuable on a long lathe where the tool may be working at some distance from the head stock.

DOUBLE HEAD FACING MACHINE.

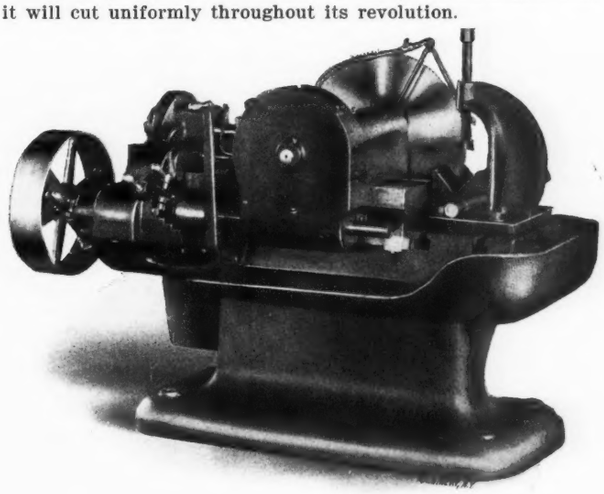
The accompanying half-tone shows a facing machine built by Fay & Scott, Dexter, Maine. It is designed to be used for facing the flanges of pipes, pipe fittings, and similar work, held in stationary fixtures on the bed. The head at the left-hand end is fixed, while that at the right is adjustable to suit the length of the work. Power is applied to the spindles on these two heads through a splined shaft, which is driven from the motor, as shown in the cut. On the nose of each spindle is mounted a star feed head, carrying a slide to which the cutter is fastened. Each head is furnished with back gears, and these, in connection with the variation obtainable in the motor, give the requisite range of speeds. The motor has a variable speed of from 470 to 1,400 revolutions per minute, and is rated at 10 horse power. This particular machine was mounted on a 26-foot bed, and its total weight was 15,000 pounds.



Fay & Scott Double Head Facing Machine.

NO. 5 COCHRANE-BLY METAL SAWING MACHINE.

This cold saw was designed with the idea in mind of giving it as great convenience, productive capacity, and neatness of design as the experience of its builders was capable of. Among the new features is the arrangement of the ways on which the carriage slides. At the saw side, where the strain is upward, the carriage is guided by a shelf solid with the main casting which bears on the underside of the way, the adjustment of wear being taken care of by a taper gib which is interposed at this point. On the opposite side another taper gib is used to take care of lost side motion. To still further increase the rigidity of the machine, the usual construction involving the use of a split box for the saw spindle has been discarded and the bearing is made solid. The saw arbor is of large size, case hardened and ground to true cylindrical form. The saw is mounted on an expansion bushing at the end of the spindle, which thus centers it, and gives assurance that it will cut uniformly throughout its revolution.



Cochrane-Bly Cold Saw.

The feed screw is made as large as practicable and placed directly in back of the saw. This screw and its nut are of the same length and the nut is split to allow all slack to be taken up. The screw is driven through a worm and wheel by a cone of gears, giving five changes of speed. An automatic stop shuts off the feed and returns the carriage for another cut. The spindle is driven by a pair of steel spur gears through a large cast-iron worm wheel and a case-hardened triple worm of coarse pitch. The driving shaft is back geared, giving a choice of two speeds to the spindle. The drip pan under the saw returns all oil to the tank at the back of the machine, whence a positive-geared pump provides an abundant supply at the cutting point. The work table is provided with T-slots for holding fixtures for special work, or the work may be strapped directly to the table and the full section of the saw outside of the collar utilized. With the machine is furnished a vise for ordinary round or square stock and a stand to support the free end of the bar. This cold saw is built by the Cochrane-Bly Co., Rochester, N. Y.

A NEW METHOD OF FINISHING CRANKSHAFTS.

The series of photographs, of which reproductions are shown below, illustrate a machine and process, for machining crankshafts, which have recently been developed. The machine itself, as may be seen by reference to Fig. 1, is an adaptation of the vertical boring mill idea. The cross rail, which is adjustable as to height above the table, carries a single slide furnished with an offset tool post extending out to the center line of the table. The rough forging comes to the machine in the shape shown at the left hand of Fig. 2. It is placed

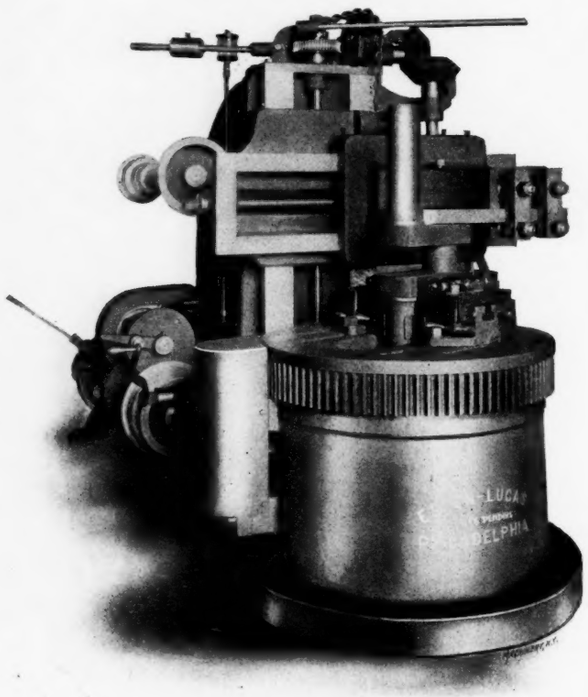


Fig. 1. Espen-Lucas Machine for Finishing Crankshafts.

in a specially constructed vise on the table of the machine, where it is grasped as shown in Fig. 3 by the lower cheek, with the lower journal extending down through the opening in the center of the table; an extended tool is clamped on the slide of the machine and used to work out the metal which in the forging fills up the throw of the crank. The advan-

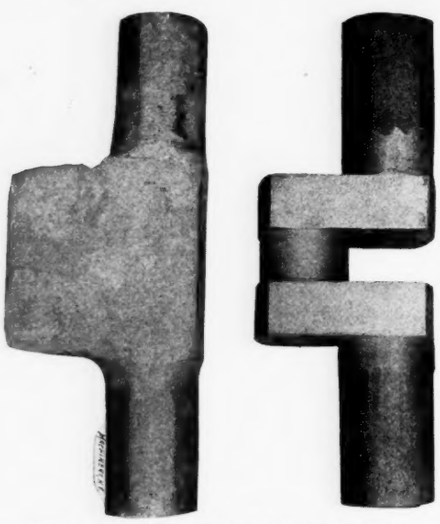


Fig. 2. The Forging and the Finished Product.

tages of performing this operation in this manner are plainly seen. The stock is held more firmly than would be possible in the lathe, since it is gripped close to the point where the cutting tool is at work. The rise and fall of the unbalanced weight of the forging which prevents rapid work when the piece is turned on centers is avoided, and, since the machine was made for this particular work, the unnecessary joints and sliding surfaces which bring down the efficiency of the lathe when performing this operation are avoided. The outside of the cheeks is roughed out in a similar manner as shown in

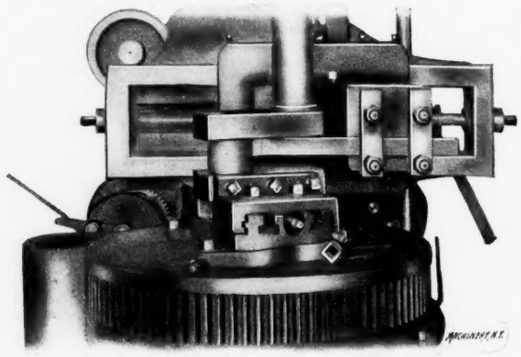


Fig. 3. Turning the Crank Pin.

Fig. 4, the vise being adjusted to bring to the center of the table, the center line of the shaft instead of the center line of the crank pin as before. A third adjustment of the vise brings the work into position to finish the ends of the cheeks.

After these cuts have been taken, the crank may be taken out of the universal vise, removed from the machine, and laid on the table to have the faces of the cheeks surfaced up, as may be dimly seen by the tool marks on the finished crank at the right of Fig. 2. The total time taken to rough out this crank from the forging was 1 hour and 30 minutes. This included cutting out the metal in the throw of the crank, turning off the ends and sides and turning up and finishing the outside of the cheeks, and finally facing off the top and bot-

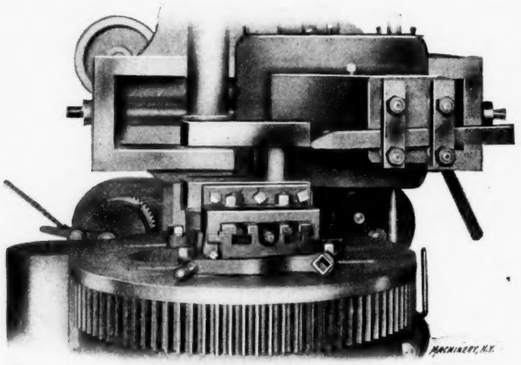


Fig. 4. Turning the Ends of the Cheeks.

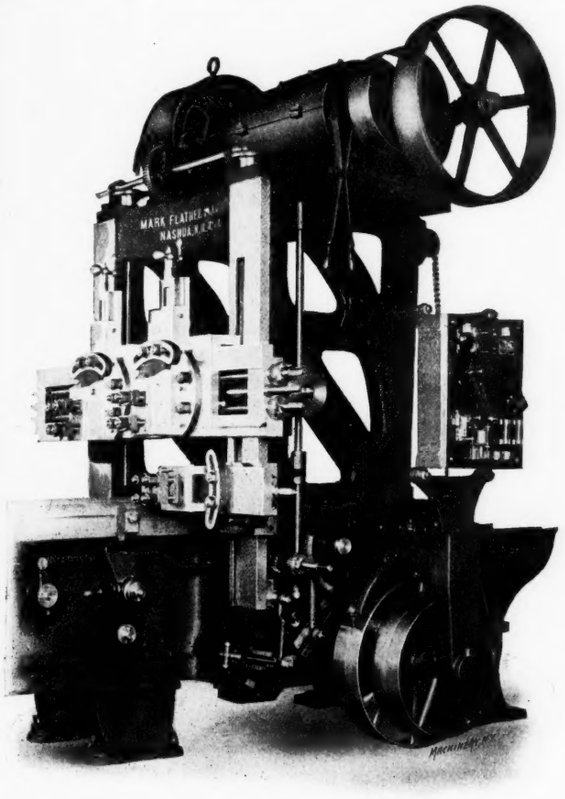
tom of the crank. This machine, with its attachments, can rough out and finish crank shafts having throws of 20 inches or smaller. The machine has automatic feeds in all directions. The gears are made of hammered crucible steel and the bearings are lined with bronze. The vise is furnished with graduations which facilitate the setting of work to varying lengths of throw.

This machine, which weighs 20,000 pounds, is built by the Espen-Lucas Machine Works, Philadelphia, Pa.

VARIABLE SPEED COUNTERSHAFT FOR PLANER.

The accompanying half-tone shows a Flather planer, built by the Mark Flather Planer Co., Nashua, N. H., fitted with a four-speed driving mechanism of recent design. The motor is directly connected with the shaft which extends through the gear box, carrying on the outer end the pulley which drives the reverse belt. This shaft, in the interior of the case, has keyed to it a cone of four gears which mesh with corresponding mating gears loosely mounted on a parallel shaft. This second shaft gives the cutting movement to the platen, through a belt and pulley mounted on its outer end. Any one of the four loosely revolving gears may be connected with the shaft on which they are mounted by throwing in a friction clutch,

of which there are four, one for each gear. These clutches are operated by the two levers which may be seen hanging down from the gear case at the side of the housing. Each lever controls two clutches, but an interlocking mechanism prevents the two from being thrown in at the same time. This arrangement thus gives a choice of four cutting speeds in combination



A Four-speed Planer Drive.

with a constant return speed. Its makers furnish it either as an attachment to the planer, or in detached shape as a counter shaft to be used with their own machine, or one of any other make.

SIXTEEN-INCH ROCKFORD SHAPER.

The Rockford Machine Tool Co., Rockford, Ill., are building the shaper shown in the cut. The column is made with overhanging ways at each end to give an increased support to the ram. The base is a pan construction for catching oil, chips, dirt, etc. It has a forward extension accurately finished for a table support. This table support is instantly adjustable to the height of the table and gives it a rigid support for heavy cuts. The ram has a screw adjustment for length of stroke and its position can be changed without leaving the work and with the machine in motion or at rest. The vise, which is made with jaws 10 inches wide and 2¼ inches deep, has an improved screw arrangement which draws the jaws together instead of pushing them, the thrust being taken by a steel collar nut, which relieves the frame of all strain tending to spring it or impair its accuracy.

The machine is driven by a 3-inch belt on a four-step cone connected to the crank through gearing, so arranged that the drive may be either straight or back-geared, the ratio being 6 to 1 for straight gearing and 19 to 1 when the back gears are engaged, with a range of from 8 to 76 strokes per minute. The cross feed is automatic in either direction. The feed-connecting rod adjusts itself to any height of rail. The elevating screw, which is of telescopic form, is provided with ball bearings and may be actuated by the same ratchet mechanism which gives the cross feed, thus providing a power vertical feed to the table. The shafting is of high carbon steel, with bearings bushed with bronze and provided with a self-oiling arrangement, fed by covered oil holes conveniently placed.

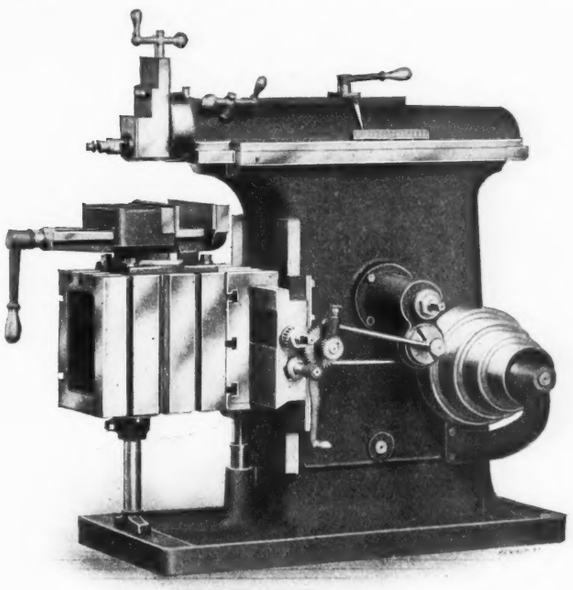
A FIVE THOUSAND POUND DROP HAMMER.

What is believed by its makers to be the largest friction board lift, drop hammer, in the world, has just been built by the Billings & Spencer Co., Hartford, Conn. It is to be used by the Bethlehem Steel Co. of South Bethlehem, Pa., in the manufacture of heavy gun forgings for government orders. The weight and dimensions of this drop hammer are in excess of those of any other tool of this class in the world.

The base weighs 72,526 pounds and the hammer itself 5,000 pounds. The uprights weigh 7,600 pounds each. The friction rolls weigh 1,200 pounds each and the roll spindles, rolls, gears and oil guards are one piece forgings. The hammer, rough planed, weighed 5,600 pounds. These forgings were made by the Bethlehem Steel Company. The shoe forging weighs 2,240 pounds and the shoe key weighs 160 pounds. The dimensions of driving pulleys are 60 inches by 13 inches by 4½ inches. The length of rear roll shaft is 94 inches; that of the front roll shaft 60 inches. The distances between the point of ways is 30 inches; the extreme fall of the hammer six feet four inches. The total weight of machine is 125,000 pounds.

* * *

Slowly but surely the wireless telegraph is finding its place, but its work is not yet the displacement of the wire telephone and telegraph lines. It rather is a valuable auxiliary to the wire lines, filling the gaps they cannot. For example: it is reported that the United States Government is preparing to install a wireless telegraph system throughout the lower Yukon Valley in Alaska, using it to replace the land lines that have proved failures on account of the difficulty of maintaining them in the face of forest fires, floods, cold weather and other obstacles. But as this region becomes more thickly settled, if it ever does, the wire lines will undoubtedly be restored for the reason that it is not probable that the wireless systems will ever be capable of the ramifications of wire



Sixteen-inch Rockford Shaper.

lines. Wireless telegraphy will find its use on the sea, the desert, and in the uninhabitable areas generally, where the maintenance of wire lines is difficult and costly. It is not to be inferred from this, however, that wireless telegraphy is cheap for, on the contrary, it is costly under the present conditions of installation and operation. The report that Professor Braun has discovered means for projecting the waves in one direction only has aroused much interest, as such a system would undoubtedly greatly extend the possible uses of wireless telegraphy, for it would reduce the cost of operation and lessen the interference in territory where a number of instruments are working simultaneously. In short, it would put wireless telegraphy more nearly on a par with the wire system in the matter of effectiveness and secrecy.

THE ECONOMY OF THE GAS PRODUCER POWER PLANT.

The two articles reproduced below are suggestive along the line of determining the conditions which limit the future fields of the various prime movers. The first discussion is taken from the October number of *Power;* the second was contributed by J. H. Kinealy to the *Engineering News*, issue of September 7.

IS GAS POWER MORE ECONOMICAL THAN WATER POWER?

That power can be turned out of a gas-power plant for less cost than from a water-power plant seems at first thought to be almost incredible, and in order to show how this may be, the writer begs to present the following analysis of costs. That we may be fair, we will take our data of the water power from a prospectus before us, detailing the cost and probable results in the development of a water power of 20,000 H. P.

From other data we find the estimate given in it compares favorably with several of the larger water powers that have been built, so it is fair to take this one as an example, and quoting from the figures therein, we have the total cost of the water-power plant at $3,555,000. This includes land, dams and canals, power station and electrical equipment, transmission lines, etc., of a plant ready for continuous operation. This is $177 per horse power.

The following estimate of the cost of operation, etc., is also taken from the prospectus:

Operating expenses per annum	$ 46,000
Interest $3,555,000 at 5 per cent	177,750
Sinking fund	100,000
Repairs and depreciation	93,000
Taxes and insurance	25,000
Total	$441,750

This plant is located at a point requiring 100 miles of transmission line.

It is perfectly fair to assume the gas-driven plant as located within this distance of a very large center of population and yet put it directly at the mine. Hundreds of mines are as near Pittsburg, Cleveland, Columbus, Indianapolis, St. Louis, Omaha, Baltimore and numerous cities of from several hundred thousand to more than a million population. So we will place our gas plant at a mine, and will assume 65 cents per ton as the price of slack coal in the hoppers of the producer. This would probably be more profitable than shipping the slack, as the producer uses mostly slack, leaving the lump for shipment.

The cost of this plant of the same capacity as the water power ready for continuous operation would not exceed $2,325,000 or $116 per horse power; consisting of gas engines, by-product producers, electrical equipment, buildings, land, transmission lines, etc., as in the case of water power ready for continuous operation. The operating costs are very nearly as follows:

Labor	$ 50,000
Interest $2,325,000	116,250
Sinking fund	75,000
Repairs and depreciation	116,250
Insurance and taxes	15,000
Total	$372,500

The plant we are proposing is located at a bituminous mine and we would only consider a by-product installation, as herein lies a great source of profit.

Every ton of bituminous coal contains from 70 to 85 pounds of sulphate of ammonia. This is worth 3 cents per pound and holds steadily at about this price. From plants in operation it is found that the average net result from the by-products is fully $1.10 per ton of coal gasified, after deducting the cost of sulphuric acid, etc., used in the process.

It is expected that the water-power plant to which we refer shall deliver power to so varied a custom that it will be practically uniform, as the peak loads of lighting and railway plants shall be taken care of by local plants and storage batteries, so that a load factor of 85 per cent may be assumed, at least for the sake of comparison.

A plant of 20,000 H. P. capacity will deliver say an average of 17,000 H. P. every day in the year, with a coal consumption of 96,798 tons. As each ton of coal nets $1.10 in by-products, this tonnage would yield $106,477. The coal bill for 96,798 tons at 65 cents is $62,918.

The net result then from the by-products is, the coal paid for and $43,559 placed to the credit of the expense account, and beside this gas sufficient to run 17,000 H. P. every day in the year.

The operating account will now stand	$372,500
Less profit on by-product	43,559
	$328,941

In order to make the comparison as nearly correct as possible, we will assume that each plant delivers 17,000 H. P. or 111,100,000 kilowatt-hours per annum and that this may be sold at 1 cent per kilowatt-hour, aggregating $1,111,000, with the following result:

	Total Receipts.	Operating Expenses.	Net Receipts.	Percentage of Profits on Investment.
Water power	$1,111,000	$441,000	$670,000	18.85
Gas power	1,111,000	328,940	782,060	33.64

From these figures it would appear that, given a coal mine and a water power equidistant from, say, Chicago, the coal mine can sell slack coal to the gas plant at 65 cents (in which there is a profit), ship the lump at an increased price, transmit current and beat the water power 14.8 per cent in results, not to consider the sale of coal direct. It will be remembered that these figures are only for the sake of comparison, as the cost of coal and the price of current will vary in every case, and the latter is put at an extremely low figure.

COMPARING THE PERFORMANCE OF GAS-PRODUCER AND BOILER PLANTS.

In the preliminary report on the operations of the coal testing plant of the Geological Survey, at St. Louis, Mo., it is stated that comparative tests of 14 bituminous coals from nine States, show the power efficiency of these coals when used in the gas-producer to be two and one-half times greater than their efficiency when used in the steam boiler plant; or, in other words, 1 ton of these coals used in the gas-producer plant has developed, on a commercial scale, as much power as 2½ tons of the same coal when used in the ordinary steam-boiler plant.

This statement means that the preliminary tests have shown that for power purposes a gas-producer plant will use only 40 per cent. of the fuel that would be used by the ordinary steam plant to develop the same power. It is well to bear in mind, while considering this statement, that what those in charge of the tests called an "ordinary steam boiler plant," was a steam plant in which the engine was a non-condensing engine of the Corliss type whose water rate was 26.3 lbs. of steam per hour per horse-power developed.

The boiler plant and the gas-producer plant were each of between 200 and 250 H. P. capacity, and we are told on page 118 of the report that the "labor required would be the same for the operation of either the boiler plant or the gas-producer plant of the capacity under tests. In either plant two men would be sufficient." This means that the difference in the costs of operation of the two plants is equal to the difference in the cost of fuel alone.

In estimating the relative economy of gas producer and boiler plants, however, account should be taken of the requirements for heating the establishment where the power is used. In general, fuel for any establishment, whether a factory or not, is used for power purposes and for heating; and the relative cost of the fuel used for each depends upon the purpose of the establishment and upon its location. For most places in the United States north of the Ohio River it is usually necessary to provide heat for heating purposes for a season of about six months. Therefore, when considering the relative cost of fuel for a steam plant and a gas-producer plant for a given establishment, we may say that, for one half of the year fuel must be used for power only, and for the other half of the year fuel must be used for power and for heating.

When the power plant is a non-condensing steam plant, the weight of fuel required for power and heating depends very largely upon the relative amounts required for each. When the quantity of steam required for power is greater than that

required for heating, the weight of fuel required for heating will be practically nothing; because by the use of a properly designed and installed heating system exhaust steam from the engine may be used for heating purposes with very little or no back pressure being brought onto the engine, and hence with little or no increase in the fuel required for power during the heating season. If the steam required for heating is greater than that required for power during the winter, then the fuel for power during the winter need not be considered, because all of the exhaust steam from the engine supplemented by live steam from the boilers, will be used for heating, and the engine will simply serve as a reducing pressure valve for such steam as passes through it before entering the heating system. Hence the fuel required during the heating season will be that required for heating only.

We see, therefore, that for the ordinary non-condensing steam plant we have two cases to consider.

(1) When the steam required for power is greater than that required for heating. Here the fuel required during the heating season is what would be required for power only.

(2) When the steam required for heating is greater than that required for power. Here the fuel required during the heating season is what would be required for heating only.

When a gas producer-plant is used for power, the weight of fuel used during the winter is the sum of that required for power and that required for heating, because there seems to be no way as yet of using the exhaust from the gas engine for heating purposes. Hence when a gas-producer plant is used for power a separate steam plant must be installed for heating purposes. This means, of course, that if the steam plant required for heating purposes is large, more men may be required than would be required for the power plant alone; and the charges for labor for the operation of a combination plant consisting of a gas-producer plant for power and a steam plant for heating may be greater than the labor charges of a steam plant for both power and heating.

* * *

DATA ON THE USE AND PERFORMANCE OF STEAM TURBINES.*

The type of apparatus to be used, whether alternating or direct current, will not materially affect the design of the station except in so far as the question of the use of reciprocating engines or steam turbines is involved. Up to this time the steam turbine, which is rapidly growing in favor for electric railway work, has been designed almost exclusively for use in connection with alternating-current generators, and the manufacturers of electrical apparatus have held out scant encouragement that its speed could ever be so modified as to make its use with direct-current generators, particularly the larger sizes, practicable. Reciprocating engines have, therefore, been regarded as the only type of steam motor available for this class of work. It is probable that this will be the case for some time to come, but it is interesting to note that considerable progress is being made in the development of direct-current turbo-generators. A number of machines of this type as large as 500 K. W. capacity are in operation, and work is well advanced upon units as large as 2,000 K. W. There seems to be good ground for the belief that this problem will be successfully solved, and that in the near future this type of apparatus will be available in sizes as large as are generally required for direct-current work.

Engineers and steam users generally have been prepared for some time to welcome any form of prime mover which could be shown to possess any considerable advantage over the reciprocating engine, as the latter had come to be regarded as having largely fulfilled its possibilities, and no very great improvement in economy was to be looked for. The steam turbine seemed to offer the solution of the question, and while, at the time of its introduction into this country, its superior economy had not been demonstrated, its great simplicity as compared with reciprocating engines, lower first cost, and less floor space occupied, insured its prompt adoption by a large number of power users, and from the first its progress has been rapid. In a report of the committee for the investigation of the steam turbine made to the National Electric Light Association last June, it was stated that there were in operation at that time 224 turbines of an aggregate capacity of over 350,000 horse power, the greater number of which had been installed in the last two years. The writer is informed that the orders for turbines taken by the largest two manufacturers in this country aggregate (July 1, 1905) over 800,000 horse power.

It is to be regretted that most of the data upon the efficiency of steam turbines has been derived from tests covering very short periods of time, usually only a few hours, and that so little data is available of their performance under actual service conditions. To the street railway manager or engineer, power station records for long periods, showing the coal consumed per kilowatt-hour, or, better still, the efficiency of the plant expressed in percentage of heat energy in the coal converted into electrical energy at the switchboard, are of much greater interest and value than the record of any number of short-time tests for steam consumption only, as it provides him with a much more practical means of making comparisons with the performance of other stations with which he is familiar. The data which has been published illustrating the relative economy in steam consumption of turbines and reciprocating engines rarely ever shows comparisons between units operating under identical conditions as to steam pressure, superheat, or vacuum and therefore does not fairly represent the relative performance of the two types, and, too, the steam consumption of the auxiliaries is also invariably omitted, so that it is impossible to form an intelligent opinion as to the additional cost of the higher vacuum required for the turbine.

Up to this time most of the turbines installed in electric railway power stations are operated in connection with reciprocating engines, and owing to the difficulty of separating the operating charges, it has been practically impossible to obtain reliable information as to their performance under commercial conditions.

One of the plants where turbines are exclusively used is the Quincy power station of the Old Colony Street Railway Company and through the courtesy of P. F. Sullivan, president of this company, the writer is permitted to publish some information regarding the performance of this station. This information was kindly furnished by C. F. Bancroft, superintendent of motive power and machinery.

It should be stated at the outset that this station, which will eventually furnish power for that portion of the Old Colony Street Railway Company's system, extending from Quincy on the north to the city of Fall River on the south, is not yet in full operation. Its connection with the latter city, where a large part of the current is to be used, has not yet been made, so that at present it furnishes power for only about one-third of the number of cars which it will eventually drive.

The station contains five 2,000 K. W., 4-stage vertical turbines, running at 750 R.P.M. and connected to 13,200-volt, 25-cycle alternating-current generators. The steam pressure is 200 pounds. There are ten horizontal water-tube boilers of 750 horse power each, equipped with internal superheaters, giving to the steam an average of 65 degrees superheat. Under-feed stokers are used. There are no economizers. One turbine is supplied with steam-driven auxiliaries; the other four have motor-driven auxiliaries. At present, while only two units are in operation, the feed water is heated to 200 deg. F. by the exhaust from the steam auxiliaries. The average daily output is 52,500 K. W. hours, giving a load factor of 54.7 per cent for the two machines. Georges Creek Cumberland coal is used, having an average calorific value of 14,000 B. T. U. per pound. The average coal consumption for this station, operating under the conditions outlined above, is 2.94 pounds per kilowatt-hour, showing an efficiency of 8.36 per cent. This record covers a period of six months, ending June 30, 1905.

While this performance does not furnish conclusive evidence of the superiority of the turbine over reciprocating engines in electric railway work, it compares favorably with the results obtained in a large number of the better class of stations using the latter type of prime movers, and gives some force

* From paper upon power plants, by Fred N. Bushnell, before the Philadelphia Convention of the American Street Railway Association.

to the opinion that in actual practice there will be found to be very little difference in the coal consumption of steam turbine and reciprocating engine plants operating under similar conditions.

* * *

FRESH FROM THE PRESS.

ELEMENTARY COURSE IN MECHANICAL DRAWING. Parts 1 and 2. By W. D. Browning, Editor of the *Draftsman*, and F. H. Sibley, Instructor of Drawing and Machine Design, Case School of Applied Science. Published by Browning Press, Cleveland, Ohio. Price 50 cents each.

The first part gives the usual directions for the selection of instruments and material, and then follow chapters on lines, lettering and figuring, geometrical problems, or orthographic projections. The second part contains chapters on development of surfaces, helices and screw threads, working drawings, lettering and blueprints.

PRACTICAL PLANER KINKS. By Carroll Ashley. 80 pages 5 x 7½ inches; 32 illustrations. Published by Hill Publishing Co., N. Y. Price $1.00.

This book is good so far as it goes, but contains too little matter to ask a shop man to pay one dollar for it. Not but that he could easily glean many times more than it cost from a careful study of its contents, but in these days of low-cost technical publications the price seems out of proportion. The cuts in many cases could have been made larger profitably, and we are sure that the writer out of his abundant experience could have added more practical matter on planer practice—a subject, by the way, on which too little has been written.

A COURSE IN TINTING. By W. B. Browning. Published by the Browning Press, Cleveland, Ohio. Price in paper, 30 cents; in cloth, 50 cents.

The book gives practical directions for tinting drawings, and five color plates showing examples of tinting. In view of the fact that some of the technical schools have introduced a small amount of such work in drawing courses, the booklet will perhaps meet with some demand in such directions. There is doubtless, however, more demand for some instructions on making wash drawings of mechanical subjects, especially in perspective from the usual three-view drawings. If a manufacturer can present to possible customers a counterfeit resemblance of what a certain machine will look like when complete, it will often result in obtaining orders before any expense is incurred in the actual construction of the machine and this, of course, is quite desirable with machines of a special nature having a limited sale. It would appear that a course along these lines would meet with favorable reception among the drafting fraternity.

ESSENTIAL DATA ON BEVEL GEARING. By Edward J. Frost. 243 pages 4½ x 5¾ inches. Published by Edward J. Frost Publishing Co., Jackson, Mich. Price $5.00.

This book is intended as a reference book and time-saver for the drafting room and shop, particularly the latter. It represents an enormous amount of personal work on the part of the author, containing upward of 70,000 calculations, which have been arranged in convenient tabulated form and which form the main part of the book. Besides the main tables, there are several other tables and diagrams in the preliminary pages. These preliminary pages are given up to brief discussions of the elements of bevel gearing and shop methods under the following heads. Relative Sizes of Teeth with Large and Small Ends, Tooth Parts Definitions, Corrected Addendum and Chordal Thickness of Tooth, Methods of Cutting and Use of the Tables. As stated, the bulk of the book, 224 pages, is occupied by the gear tables for pinions and gears at right angles, beginning with 9-tooth pinion table, and concluding with a 70-tooth pinion table. Each table gives the face angle of the pinion, cutting angle, outside diameter for 1 pitch, backing for 1 pitch, number of cutter to use, back angle and pitch cone radius. The back angle and cone radius are the same for both pinion and gear. The face angle, the cutting angle, outside diameter, backing and number of cutter are also given for the mating gear in each case. The first table starts with a 9-tooth pinion and gives this data for it and mating gears up to 132 teeth by intervals of one tooth. The following table gives the same data for a 10-tooth pinion and mating gears up to 132 teeth, and so on up to a pinion of 70 teeth, each table giving the elements of both pinion and gear up to 132 teeth. It is obvious that a reliable book of this kind will be most valuable to the busy shop superintenent or foreman as it gives the necessary data for turning and cutting almost all bevel gear blanks likely to be met with in practice. Of course this data cannot be used directly, except in the case of the angles and gears of 1 pitch, but the simplicity of the calculations necessary to change the data to any required pitch makes them easily mental processes in most cases. Each table starts, of course, with the number of teeth by which it is designated so that each table is one place shorter than the preceding one, but the data for all pinions and gears within the limits of numbers named is included. It would be especially valuable on repair work, and should save its cost many times over where much of such work is being done. Being of a convenient pocket size, it can be carried on the person and referred to whenever required. Mr. Frost cannot claim absolute accuracy for the tables, but he has carefully checked all calculations and believes that they are, with very few exceptions, correct.

NEW TRADE LITERATURE.

E. G. SMITH, Columbus, Pa., catalogue giving list of all sizes and styles of Columbia calipers, and describing the special advantages of the individual types.

NORTHERN ELECTRICAL MFG. CO., Madison, Wis. A leaflet describing and illustrating Northern forge blowers. A table giving the general details and an idea of the capacity, etc., is appended.

THE CRANE CO., Chicago, Ill. Circular No. 295 E of Craneweld Flanged Pipe Joints describing the process of making and giving dimensions and prices. Illustrated.

THE B. F. STURTEVANT CO., Hyde Park, Mass., have sent to their friends a Boston time table with an invitation to visit them at their plant.

AMERICAN PULLEY CO., 29th and Bristol Sts., Philadelphia, Pa. A leaflet containing an extract from a report of tests of the American pulley made at Cornell University.

FOOS GAS ENGINE CO., Springfield, Ohio. A catalogue of Foos Gas and Gasoline Engines. A general description is followed by some remarks on the important features, and illustrations of the engines.

NORTHERN ELECTRICAL MFG. CO., Madison, Wis. Bulletin No. 51 describing the construction of Northern Ring Type machines. Details of special parts are given and the bulletin is well illustrated throughout.

ARMSTRONG MFG. CO., Bridgeport, Conn. New catalogue of Armstrong Stocks and Dies, stating the points of superiority of these over other stocks and dies and describing their various types. An index is given in the front.

THE NATIONAL MACHINE TOOL CO., Cincinnati, O., have published a booklet entitled "Busy Men's Opinions," containing letters of recommendation for their key-seating tools. These letters are written by firms who have used the tools and have found them to give entire satisfaction.

THE R. D. NUTTALL CO., Pittsburg, Pa. Catalogue describing briefly their plant and products. It contains many illustrations. The new gear cutting machine which they have lately installed allows the cutting of gears up to 20 feet in diameter, 3 feet face and 7 inches p tooth.

THREE RIVERS TOOL CO., Three Rivers, Mich. Circular of the Mathews Core Drills and Reamers, with high-speed steel inserted blades, which render the tools more efficient than when made of carbon steel, and much less expensive than if made entirely of high-speed steel.

THE BLAISDELL MACHINERY CO., Bradford, Pa. Bulletin 15, Type A Compressors gives tables with specifications of simple and duplex steam-driven and belt-driven air compressors. It is well illustrated with half-tones showing the machines as wholes and their special parts.

BAUSH MACHINE TOOL CO., Springfield, Mass., have issued an attractive catalogue on Drilling Machines. The interesting introduction describes the adjustable multiple spindle drills and invites attention to the illustrations of the machines which they have made. A list of firms using their machines is also given.

THE CLING-SURFACE CO., Buffalo, N. Y., have issued a leaflet containing letters from the Broadhead Worsted Mills and the Consolidated Lamp & Glass Co., which testify to the excellence of Cling-Surface as used on their belts. By means of this, slack-running belts may be used and the extra expense in time and money of overcoming friction is done away with.

THE GISHOLT MACHINE CO., Madison, Wis., have sent out an attractive and original catalogue in the form of a booklet called "Don'ts for Machinists." The list of Don'ts as published in MACHINERY occupies several pages, while the few remaining pages are devoted to Gisholt turret lathes, boring mills, horizontal mills and tool grinder. This is an interesting job of printing.

THE CROCKER-WHEELER CO., Ampere, N. J. Flyer No. 276, entitled Automatic Belt-Tightening Idler described a newly-designed automatic belt-tightening attachment for the standard Crocker-Wheeler Form L motor. It may be used wherever the limited center distances between pulleys require an increased belt contact on the pulley surfaces, as on the Lanston Monotype machine.

THE INGERSOLL-SERGEANT DRILL CO., 11 Broadway, New York City. Form 52 A on Coal Mining Machinery, giving a brief description of boring machines, coal cutters, etc., and referring to preceding publications which contain complete information on these topics. Also Bulletin No. 2004 on Stone Working Tools. Some interesting examples of architectural stone cutting are shown.

PHILADELPHIA GEAR WORKS, INC., Philadelphia, Pa. A comprehensive Gear catalogue containing lists of the varieties of gears. This catalogue also contains other interesting information in the form of rules for sizing gears, table of tooth parts, bevel gear chart, and description of the method of drawing the bevel gear, spur gear wheel, and standard involute tooth.

THE AMERICAN BLOWER CO., Detroit, Mich. Catalogue of Heating Apparatus. Interesting descriptions of the various types of heating and ventilating fans are given and their special uses are set forth. A system of ventilating a school room is described and illustrated. With this catalogue is a list of specifications of their Vertical High Speed Automatic Engine and a treatise on the relative cost and return of their fans and blowers.

MINER & PECK MFG. CO., New Haven, Conn. Catalogue of Peck Automatic Drop Lifter. By means of this lifter the hammer is brought from a state of rest gradually instead of being jerked up suddenly, thus securing uniform blows. The New Model Lifter has the weight of the parts removed from the lifting shaft and a heavier blow is thus secured. Their drop presses, blast forges, and wind gates are also described and illustrated.

BALDWIN LOCOMOTIVE WORKS, Philadelphia, Pa. Catalogue No. 50, Electric Trucks, describing and illustrating their recent electric trucks made for various railway companies. Also Catalogue No. 53, containing notes on the balanced compound locomotive taken principally from a paper on this subject read before the Richmond Railroad Club, and from articles published by various technical journals. The different types are illustrated and described.

W. P. DAVIS MACHINE CO., Rochester, N. Y. Catalogue F of machine tools, engines, boilers, and supplies. This is a cloth-bound book of 400 pages, listing the large line of machinery and tools carried by this firm. This company manufactures all sizes of lathes from a foot-power lathe to a lathe of 42 inches swing, besides upright drills, key seaters and other tools. They also handle tools of a number of prominent firms, besides a large line of engines, boilers, factory and mill supplies, etc. The catalogue is for distribution among purchasers of machinery who are interested.

THE R. K. LEBLOND MACHINE TOOL CO., Cincinnati, O. Catalogue of milling machines. A general description of milling machines is given first which is followed by descriptions of special types. Also two catalogues on lathes. The same general plan is followed in these as in the milling machine catalogue—beginning with a general description of lathes, and going on to specifications of the individual types. One of the lathe catalogues is an expansion of the other, giving more types, and a little more detail of description. The catalogues are well illustrated throughout.

MANUFACTURERS' NOTES.

THE INGERSOLL-RAND CO., 11 Broadway, New York City, have established a branch office at Houghton, Mich., under the management of Mr. T. F. Lynch. A complete stock of repair and duplicate parts for all Ingersoll-Rand machinery will be carried at the new office.

THE WILMARTH & MORMAN CO., Grand Rapids, Mich., have been obliged to secure larger quarters, owing to the rapid increase of their business. They are now located near the corner of Canal and Leonard Sts., where they have secured a long-time lease with option of purchase.

THE S. OBERMAYER CO., Cincinnati, O., through their New York representative, have secured the contract for a complete brass foundry equipment for the brass foundry to be installed for the fire department of New York City. This will be a complete repair shop for the metropolis fire department.

THE AMERICAN BLOWER CO., Detroit, Mich., have large new contracts on hand. They are to provide the complete heating apparatus for the new Allegheny shops of the Pennsylvania lines west of Pittsburg, and the ventilating outfit for the Altman building, New York, and the Marshall building, Chicago.

THE NORTHERN ELECTRICAL MFG. CO., Madison, Wis., offer to send their purchasers quotations on Northern machines for application to machines and machine tool drive upon learning of the class of work to be accomplished and the voltage of the power circuit from which the current is to be taken.

THE WAGNER ELECTRIC CO., St. Louis, Mo., manufacturers of alternating and direct-current motors, also electrical instruments, will break ground in November for their new factory, just outside the western limits of the city. The buildings will cover twelve to fifteen acres.